PROTOTYPE DESIGN, TESTING, AND ANALYSIS

Component Selection and Design Refinement Report

Third-Generation Prototype Fabrication and Laboratory Test Report

TECHNICAL REPORT

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Acknowledgements

Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to $62 million to conduct the most promising public interest energy research by partnering with Research, Development, and Demonstration (RD&D) organizations, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following six RD&D program areas:

- Buildings End-Use Energy Efficiency

- Industrial/Agricultural/Water End-Use Energy Efficiency

- Renewable Energy

- Environmentally-Preferred Advanced Generation

- Energy-Related Environmental Research

- Energy Systems Integration

What follows is an attachment to the final report for the Design Refinement and Demonstration of a Market-Optimized Residential Heat-Pump Water Heater project, Contract Number 500-98-028, conducted by TIAX. This project contributes to the PIER Building End-Use Energy Efficiency program.

This attachment, "Prototype Design, Fabrication, and Testing" (Attachment A-1), provides supplemental information to the project's final report and includes the following reports:

- *Component Selection and Design Refinement Report*

- *Third-Generation Prototype Fabrication and Laboratory Test Report*

For more information on the PIER Program, please visit the Commission's Web site at: http://www.energy.ca.gov/research/index.html or contact the Commission's Publications Unit at 916-654-5200.

Abstract

This "Prototype Design, Fabrication, and Testing" attachment is a set of reports produced by the Design Refinement and Demonstration of a Market-Optimized Residential Heat-Pump Water Heater project, funded by the California Energy Commission's Public Interest Energy Research (PIER) Program.

"Market-optimized" means that cost and performance are balanced to meet market needs. Specifically, the market-optimized heat-pump water heater (HPWH) is less expensive and easier to install relative to other HPWHs on the market. Prior to this project, with support from the U.S. Department of Energy Oak Ridge National Laboratory, TIAX and EMI developed and tested two generations of prototype market-optimized HPWHs. The overall scope of this project was to further refine the design of the market-optimized HPWH and then verify the third-generation design through both laboratory and field testing. Oak Ridge National Laboratory conducted laboratory-based durability tests on ten units, simulating the equivalent of ten years of heat-pump cycling under normal operation. TIAX and EMI then conducted a yearlong field test in California on 20 units.

This attachment, "Prototype Design, Fabrication, and Testing" (Attachment A-1), provides supplemental information to the project's final report and includes the following reports:

Component Selection and Design Refinement Report
This report outlines key elements of the third-generation HPWH design.

Third-Generation Prototype Fabrication and Laboratory Test Report
Contains laboratory test results on the third generation prototype unit.

Attachment 1

Component Selection and Design Refinement

Attachment to Final Report: Design Refinement and Demonstration of Market-Optimized Heat Pump Water Heater

April 2004

Note: This attachment is ***NOT*** a stand-alone report. It should be interpreted only in the context of the final report from which it is referred.

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1.0   Design Approach

Significant previous design and testing led to the second-generation prototype design. While additional design refinements are needed, the second-generation prototype provided a strong starting point. The contract defined the components for which a design/selection re-evaluation was desirable. As mentioned above, we added to this list selection of the HPWH control system (conventional versus micro-controller based).

The design approach is iterative. Each component in the system impacts the performance of other system components. Fortunately, the previous design and testing efforts allowed us to make an educated selection of design conditions, reducing the number of iterations required under this design and selection effort.

We utilized multiple design tools in the component design and selection process, including:

- Performance simulation using semi-empirical techniques;

- Matrix-type design evaluations (trade-off studies) using weighting factors and relative ratings of various performance attributes (providing a numerical comparison of design options); and

- Laboratory testing to establish empirical inputs for performance simulations, or to verify the results of design evaluations.

We applied the appropriate mix of tools for design and selection of each component. Not all tools were applied to all component designs.

We investigated all major components and subsystems of the HPWH, with the exception of the storage tank and the evaporator. The second-generation design utilizes a storage tank manufactured for conventional electric water heaters. This provides a compatible footprint, low first cost, and proven performance. We know of no other storage tank design options having comparable characteristics. The evaporator used in the second-generation prototype provided a good tradeoff between performance, physical size and shape, and reasonable cost characteristics. We could identify no significant evaporator design/selection refinements needed.

We placed high priority on achieving design simplicity, low cost, and reliability, while maintaining good energy efficiency.

As mentioned above, previous design and testing work (funded both by Arthur D. Little and DOE/Oak Ridge National Laboratory), which led to the second-generation prototype, provided a strong starting point for design refinements. Table 1-1 lists the key components used in the second-generation prototype, which establishes the starting point for the current component design and selection process. Table 1-2 shows the previous estimated manufactured cost for the market-optimized HPWH, based on a production volume of 50,000 units/year by one manufacturer.

Table 1-1: Components Used in Second-Generation Prototype

Component	Description
Compressor	Danfoss Model FF7.5GK, 120 VAC
Condenser	¼" OD x 88-ft. long copper tube, followed by 3/16" OD x 12-ft. long copper tube subcooling section; condenser spirally wound around and solder bonded to the storage tank
Expansion Device	Capillary Tube, 0.049 ID by 5-ft. long, with Suction Line Accumulator
Evaporator	Heatcraft, Nominal 2500 Btu Capacity
Evaporator Fan	Single, fixed-speed fan, 250 cfm (nominal)
Condensate Management System	Electric-resistance immersion heater (350 Watt), controlled by float switch, and installed in a separate reservoir (not in condensate pan)
Storage Tank	American Water Heaters, 50-gallon nominal (45-gallon actual) water heater tank
System Controls	Conventional electric water heater controls

Table 1-2: Previous Estimated Manufactured Cost for the Market-Optimized HPWH

Component	Cost
Compressor	$45.00
Condenser (1/4" by 100 ft. copper tubing)	$16.00
Evaporator and Fan Assembly	$25.00
Capillary Tube	$1.00
Condensate Management System	$10.00
Storage Tank Assembly (50 gallon)	$100.00
Miscellaneous Components	$10.00
Material Subtotal	$207.00
Assembly Labor	$25.00
Total Manufactured Cost	$232.00
Based on production of 50,000 units/year by one manufacturer.	

2.0   Design Conditions

The HPWH should operate effectively in virtually any environment in which electric-resistance water heaters are currently installed (except unsheltered, outdoor installations). When conditions are not appropriate for operation of the heat pump itself, the HPWH must utilize electric-resistance back-up to provide water heating. The design conditions implied by this requirement are described below.

2.1. Environmental Conditions

We considered environmental conditions experienced throughout California and the entire U.S. While the focus of this project is on developing a HPWH suitable for California, the HPWH must also be suitable for installations throughout the U.S. (We're assuming that no manufacturer will commercialize a HPWH that is not suitable for use throughout the U.S.)

Nationally, storage water heaters are installed in various locations within the residence, such as:

- Utility closets;

- Unenclosed areas in the conditioned living space;

- Basements;

- Attics; and

- Garages or other unheated shelters.

Some water heaters are installed in unsheltered, outdoor locations. However, this type of installation is uncommon, and requires a specially designed water heater. Therefore, we have not considered designing the HPWH for unsheltered, outdoor installations.

To avoid potential pipe freezing, water heaters are generally not installed in unheated areas in climates having winter temperatures substantially below the freezing point.

For greatest market impact, the HPWH should operate effectively in most, if not all, of the above installation locations. Unfortunately, hard data are not available on the fraction of water heaters installed in the various installation locations. We focused on the following representative installation locations:

- Enclosed, conditioned space (utility closet);

- Unenclosed conditioned space (open living space or basement); and

- Unheated space (garage or other shelter).

Installations in Enclosed, Conditioned Spaces

Installations in enclosures within the conditioned living space (i.e., utility closets) are very common. Section 3.1 discusses the implications of heat-pump heating capacity on enclosure temperature. We limited the capacity of the heat pump to ensure that the air temperature in the enclosure does not fall below 55°F (corresponding to an estimated maximum 80 percent relative humidity).

Installations in Unenclosed, Conditioned Spaces

For design purposes, we assumed water heaters installed in unenclosed, conditioned spaces are generally exposed to 70°F/50% relative humidity (RH) air. While indoor air conditions vary, the variance is generally modest, and is not of particular concern from a design perspective.

Installations in Unheated Spaces

While most electric water heaters are installed in conditioned spaces, we must consider installation in unheated spaces such as garages and other unheated shelters, where the water heater is exposed to outdoor temperature swings. While the shelter itself may moderate the temperature swings, we assumed (worst-case scenario) that the water heater would be exposed to outdoor temperature conditions. Andersson, et. al.[1] developed an eleven climate-region breakdown of the U.S. The cities representing each of those regions are New York, Detroit, Los Angeles, Houston, Atlanta, Dallas, Minneapolis, Seattle, Fresno, Denver, and Phoenix. We assumed that water heaters in New York, Detroit, Minneapolis, and Denver are generally installed in heated spaces (for freeze protection). For the remaining cities, Figure 2 -1 and Figure 2-2 show the percentage of time the ambient is above selected temperatures and below selected temperatures, respectively. These data are useful in establishing the desired heat-pump operating envelope (discussed in Section 2.2 below).
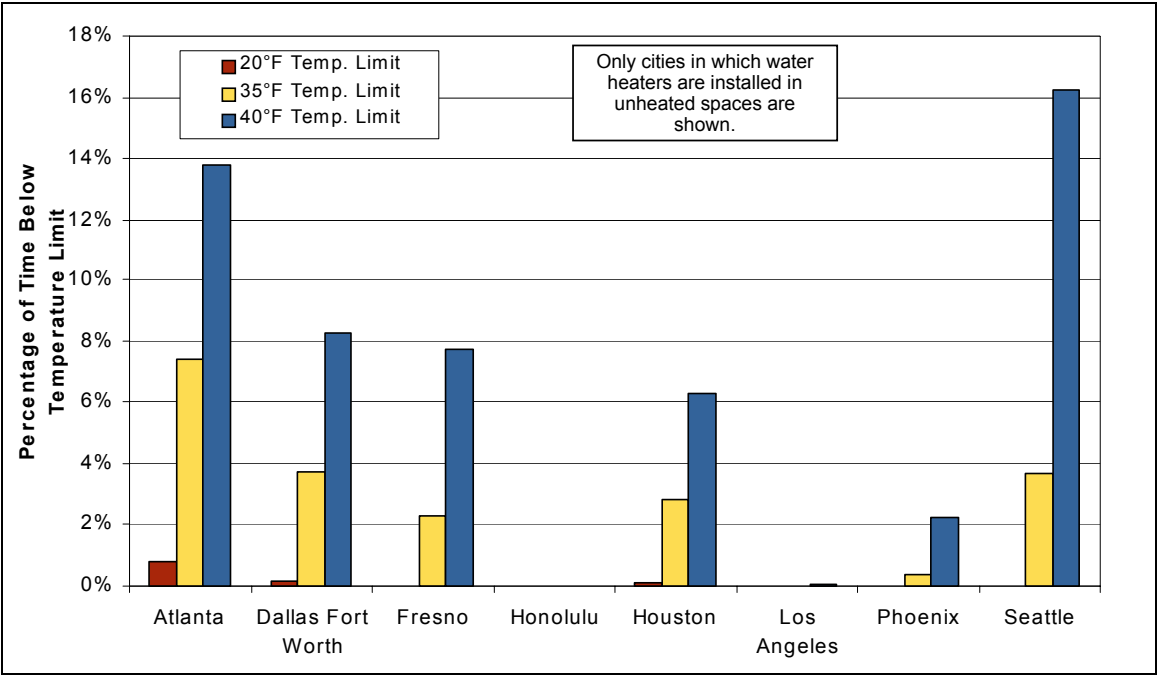
Figure 2-1: Low-Ambient Temperature Conditions for Representitive Cities

Figure 2-2: High-Ambient Temperature Conditions for Representative Cities

We also selected six California cities representing the full range of environmental conditions experienced in the state (see Figure 2-3). Where warranted for a particular component or subsystem design, we also investigated operation in these California cities.

The lowest water supply temperature experienced in the U.S. 35°F Therefore, we set the minimum supply water temperature at 35°F.

2.2. Heat-Pump Operating Conditions

Based on the environmental conditions discussed above and certain heat-pump operating constraints, we applied a common set of heat-pump operating conditions and assumptions to the design and selection of each component:

Figure 2-3: Representative California Cities

- A nominal heating capacity (at 90°F average tank water temperature and 70°F air temperature) of 3600 Btuh;

- One third of compressor input power is lost through convection and thermal radiation from the compressor shell and piping to the condenser;

- Minimum air temperature of 40°F;

- Maximum air temperature of 90°F;

- Worst-case humidity condition of 67°F dry-bulb and 59°F wet-bulb temperatures (about 85 percent RH);

- Water-draw profile from the DOE Energy Factor test procedure (64.2 gallons/day, drawn in six equal draws over a 6-hour period);

- Minimum water temperature of 35°F;

- Maximum water temperature of 135°F; and

- Operation on single-phase, 240 VAC power – maximum power draw of 4500 W.

The above operating conditions apply to the *heat-pump* only. As discussed previously, the HPWH will operate under any conditions an electric-resistance water heater operates. When conditions are outside the heat-pump operating range, the HPWH can utilize backup resistance heat.

The nominal heating capacity requirement is discussed further in Section 3.1.

The assumed compressor power loss is a typical rule of thumb applied to heat-pump design. This loss, while difficult to measure directly, is important to account for when estimating heating capacity and energy efficiency.

As discussed above, most installations are in conditioned spaces and see temperatures well within the range of 40°F and 90°F. For outdoor conditions, Figure 2-4 shows the air-temperature operating envelope overlayed on the outdoor design temperature ranges for the cities representing outdoor installations across the nation. When outdoor temperatures are outside of this envelope, the HPWH can operate on resistance backup heat.
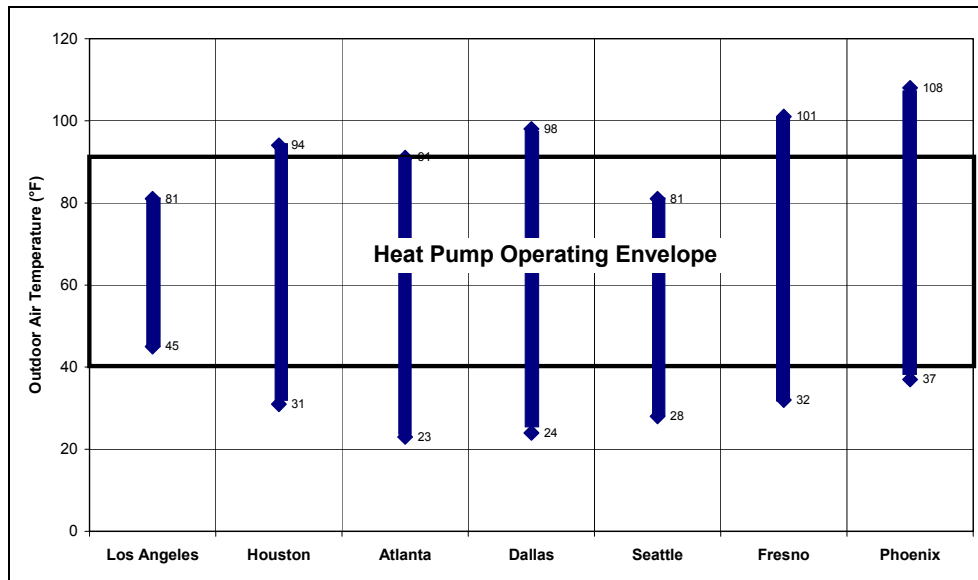
Source: ASHRAE 1997 Fundamentals

Figure 2-4: Heat-Pump Operating Envelope

The worst-case humidity condition corresponds to worst-case outdoor humidity design conditions (Eureka, CA). Humidity conditions in an enclosure are less severe than this condition.

Water-draw profiles can vary tremendously – there is no definitive profile that can be used for water heater design. The DOE test procedure, however, provides a reasonable representation of a draw profile. The daily draw (64.2 gallons/day) called out by the DOE test procedure is actually higher than average consumption in the U.S., but is probably representative of households that would benefit from using a HPWH (households having above-average water consumption will generally be more economically attractive for HPWH installations).

The 35°F minimum water temperature corresponds to the minimum supply water temperature experienced in the U.S. (as discussed in Section 2.1 above).

The 135°F maximum water temperature was established based on previous HPWH testing experience. Operating the heat pump at tank temperatures significantly above 135°F can damage the heat pump (the actual temperature limit depends on the HPWH system design). High water temperatures require high compressor discharge pressures, which result in high compressor discharge temperatures. If discharge temperatures exceed about 220°F, the polyol-ester lubricant used in the compressor can break down. The thermostats of all water heaters delivered in the U.S. are factory set at 120°F, as required by law. In reality, end users rarely change this set point. In any case, the 135°F maximum temperature applies to

the *heat-pump* only – not the HPWH. The HPWH can use electric-resistance backup to deliver water exceeding 135°F.

Residential electric storage water heaters used in the U.S. generally operate on single-phase, 240-VAC, 20-amp service. While, in theory, this means the water heater could draw 4800 W (nominal), some households have "soft" circuits (i.e., circuits that can't handle the full rated load). Conventional electric-resistance water heaters draw no more than 4500 W (nominal), corresponding to the operation of one tank heating element. To avoid an increased tendency to trip circuit breakers or blow fuses, the HPWH must also draw no more than 4500 W (nominal).

3.0   Component Selection and Design Refinement

3.1. Compressor Selection

One of the design philosophies behind the development of the market-optimized HPWH is to limit the water heating capacity. There are a number of benefits to this approach:

- A compressor with a similar size to that of a domestic refrigerator compressor can be selected and satisfy the capacity requirements for the heat-pump cycle, reducing the overall system cost and increasing system reliability;

- All of the system components are smaller and can be integrated into the physical envelope of a conventional, electric water heater without a significant height increase;

- The HPWH can be installed in an enclosure (utility closet) without danger of chilling the enclosure to an unacceptable level; and

- It limits condensate generation rate.

The properly sized compressor balances the trade-off between the evaporating capacity and the condensing capacity.

The evaporating capacity directly effects;

- The space-cooling impact;

- The condensate generation rate;

- The size (and power draw) of the evaporator fan; and

- The physical size of the evaporating coil.

As the evaporating cooling capacity increases, the space-cooling impact also increases. It is a positive impact during the cooling seasons, but it is a negative impact during the heating seasons. In addition, as the cooling capacity increases, it is likely that the need for latent heat removal is also increased, in turn increasing the condensate generation rate.

The condensing capacity primarily effects:

- The water heating capacity of the HPWH;

- The efficiency of the HPWH;

- The number of hot-water run-outs; and

- The physical design of the condenser.

The lower the water heating capacity of the heat-pump, the more the HPWH will rely on the electric resistance elements to provide hot water, decreasing the overall efficiency and increasing the likelihood of hot water run-outs. However, as the heating capacity is increased, the physical design of the condenser is affected, i.e., it increases:

- The length of the tube;

- The pressure drop through the condenser;

- The charge volume of the refrigerant; and

- The maximum temperature difference between the condenser and the water in the tank.

3.1.1. Compressor Evaporating Capacity

The space-cooling impact is greatest within the enclosed space, i.e., a closet installation. Therefore, the evaporating capacity of the compressor is selected so it is less than or equal to the heat gain within the closet. The sources for heat gain within the closet: include; heat gain through the closet wall, heat loss from the water tank, heat loss from the compressor shell, and an air-exchange with the exterior conditioned space.

The heat gain for a closet of typical construction[2] was calculated for interior temperatures ranging from 40°F to 60°F and is shown in Figure 3-1 In addition, the corresponding condensing capacity[3] is calculated determine in order to calculate the heat loss from the tank (10% of the heating-capacity) and the heat loss from the compressor shell (1/3 the power input). As is observed in Figure 3-1, as the closet temperature decreases, the temperature difference between the exterior and interior increases. The heat gain of the closet therefore, also increases.

110°F Condensing Temperature
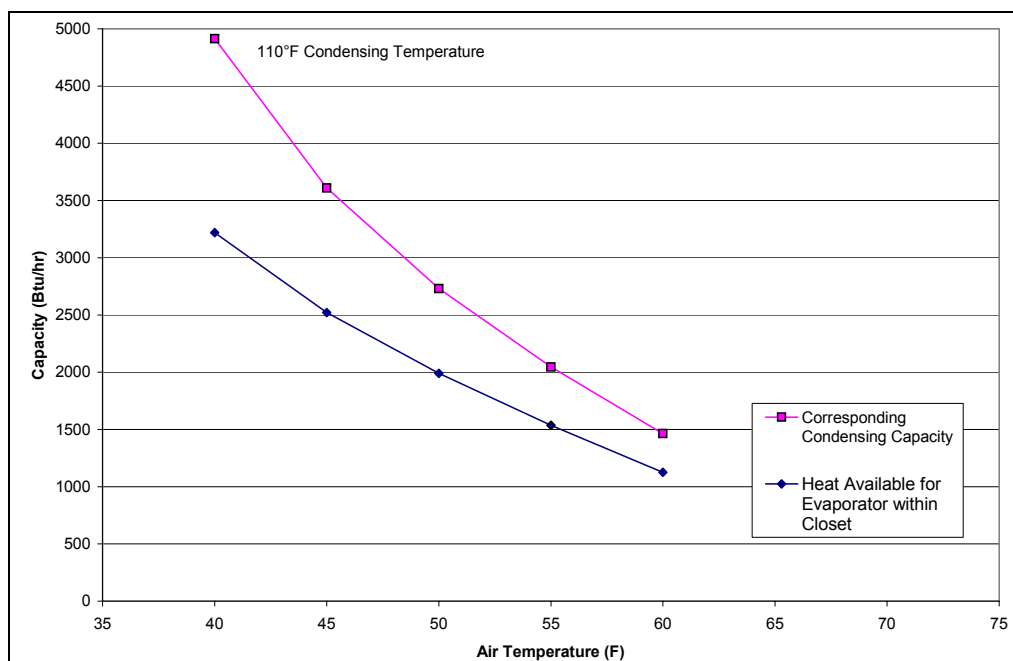
Figure 3-1: Heat Gain Within Closet Interior

To avoid condensation on the exterior walls of the closet, the minimum allowable temperature within the closet is dependent on the dew point temperature of the living space surrounding the closet. By attempting to keep the interior closet temperature above 50°F, the likelihood of condensate forming on the exterior walls is substantially reduced (refer to Table 3-1).

Table 3-1: Dewpoint Temperatures for Closet Installations

Typical Exterior Temperature (°F)	Typical Exterior RH (%)	Corresponding Dewpoint Temperature (°F)
65	50	46.0
70	50	50.5
75	50	55.0

3.1.2. Condenser Capacity

During a DOE 24-Hour Test (the test used to measure the Energy Factor), there are six water draws (10.3 gallons each) taken at one-hour increments (for a total water draw of 64.3 gallons) (see Figure 3-2). If the heating capacity of the compressor is not sufficient to heat the incoming water to a minimum of 118°F (135°F upper thermostat setpoint minus a 17°F deadband) the upper element will energize and negatively impact the Energy Factor rating of the HPWH.

We developed a computer model of the condenser based on past experimental work (refer to Appendix I) in an effort to predict the required heating capacity of the compressor. Using the model, we estimated the heat input required to maintain a water temperature of 118°F at the upper thermostat was approximately 3950 Btu/hr. With this heating capacity, the model of the HPWH was able to fully recover without the use of the upper resistance element.

Water Temperatures legend: 135 +, 135, 118, 90 +, 80 +, 60 +, 58

Tank Ht.	First Draw (t = 0 hr)	Second Draw (t = 1 hr)	Third Draw (t = 2 hr)	Fourth Draw (t = 3 hr)	Fifth Draw (t = 4 hr)	Sixth Draw (t = 5 hr)
40 - 45	135	135	135	135	118	118
35 - 40	135	135	135	118	118	118
30 - 35	135 (Upper T-Stat)	135	135	118	118	118
25 - 30	135	135	118	118	118	95
20 - 25	135	135	118	118	95	90
15 - 20	135	118	0.30 / 90	0.35 / 86	0.35 / 86	0.35 / 86
10 - 15	135	0.30 / 90	0.30 / 90	0.30 / 67	0.30 / 62	0.30 / 62
5 - 10	0.65 / 58	0.35 / 58	0.30 / 58	0.30 / 58	0.30 / 58	0.30 / 58
0 - 5	0.35 / 58	0.35 / 58	0.10 / 58	0.05 / 58	0.05 / 58	0.05 / 58

Normalized Heat Flux: 1.0, 0.5, 0.0

Figure 3-2: Simulated Tank Temperatures During the Water Draw Portion of a DOE 24-Hour Test

The DOE 24-Hour Test is performed with an ambient condition of 70°F and 50% RH. Using the 70°F as the base design point (with a 110°F condensing temperature and a 40°F evaporating temperature), we evaluated the condensing capacity at lower ambient temperatures (refer to Figure 3-3).
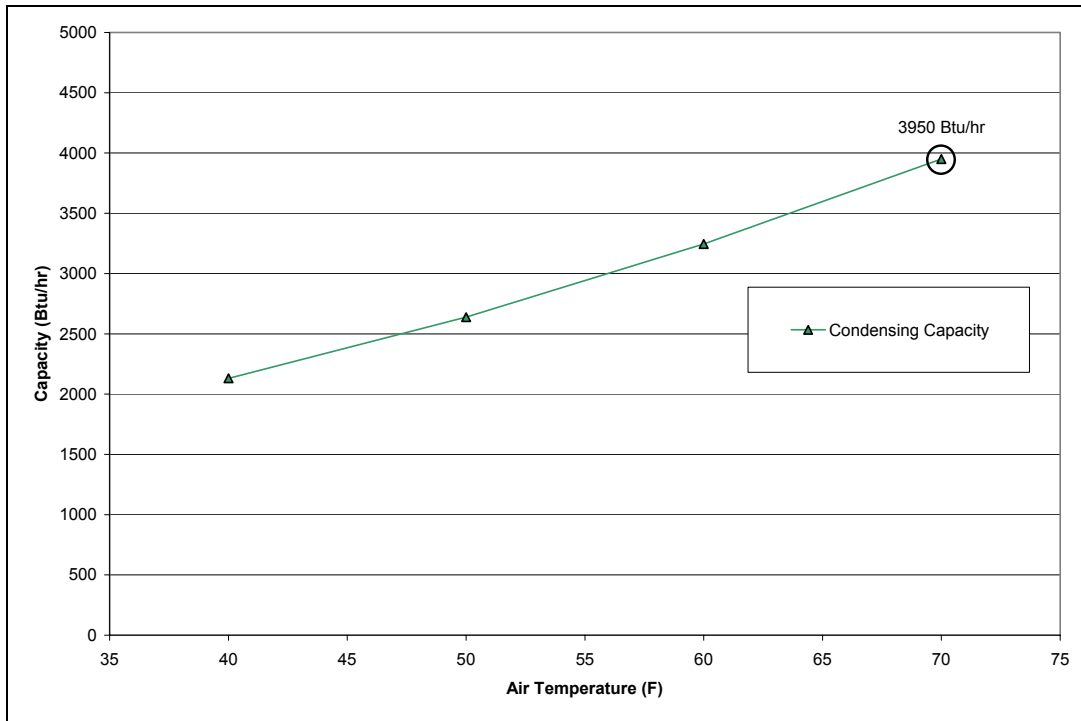
Figure 3-3: Condensing Capacity as a Function of Ambient Temperature

3.1.3. Ideal Compressor Size

Figure 3-4 combines the results from the study of the maximum evaporating capacity with the results for the minimum condensing capacity results. The ideal compressor size is located at the intersection of the two condensing capacities. This is the balance-point between the maximum evaporating capacity and the minimum condensing temperature. The minimum ambient temperature is greater than 50°F (although just slightly) and the condensing capacity is sufficient to prevent the upper element from energizing during the DOE 24-Hour Test.
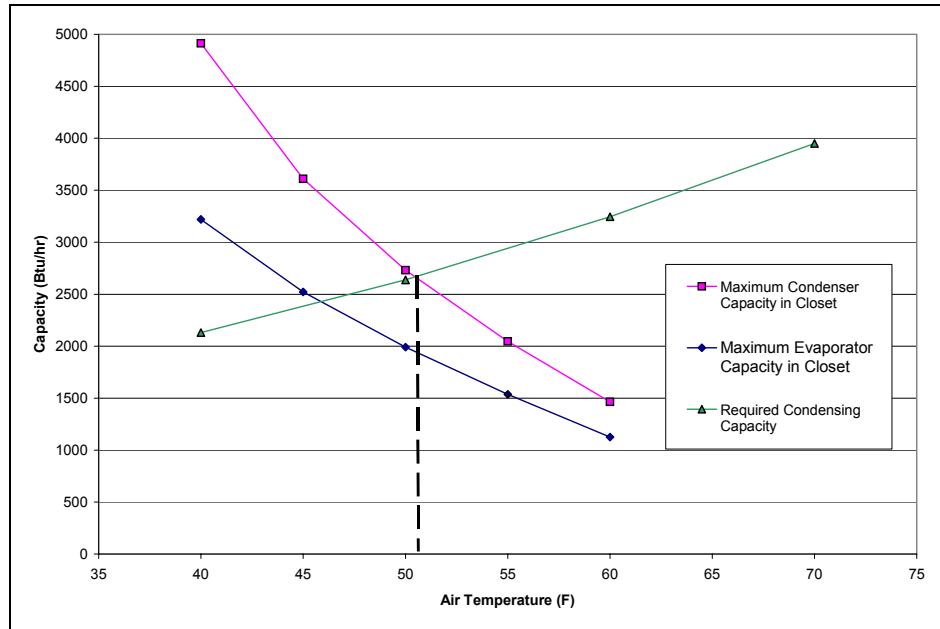
Figure 3-4: Ideal Compressor Size

3.1.4. Compressor Selection

A number of manufacturer's compressors potentially match the capacity requirements determined from this study. However, our manufacturing partner has an existing relationship Embraco Manufacturers. For this reason, we have selected a compressor with the characteristics listed in Table 3-2.

Table 3-2 Selected Compressor Characteristics

Manufacturer	Embraco
Model No.	FF10HBK
Displacement	9.05 cm^3 (0.55 in^3)
ASHRAE HBP Capacity (Cooling)	3530 Btu/hr (40°F / 130°F)
Max. Refrigerant Condensing Temperature	150°F
Max. Refrigerant Evaporating Temperature	50°F
Power	240 V / 1 Phase / 60 Hz

3.2. Condenser Design

The condenser design plays an important role in the reliability, cost, and performance of the HPWH. The consideration of a wide-assortment of trade-offs is necessary during the selection of the condenser. Specifically, the design impacts:

- The condenser length (cost),

- The charge-volume of refrigerant (cost and reliability),

- The pressure drop through the condenser (increased discharge temperatures – reliability and performance), and

- The temperature difference between the condenser and the water in the tank. (increased condensing temperatures – performance and reliability).

3.2.1. Second-Generation Prototype Condenser

The second generation prototype of the HPWH used approximately a 100 ft. of ¼ in. copper tubing for the condenser. The tubing was wrapped around the exterior of the water tank and soldered in place. The condenser's de-superheating section began nearly 16 in. from the top of the tank. The condenser continued from the de-superheat section (at approximately 1 in. spacing between wraps) until nearly 5 in. from the bottom.

Although the solder bond offers small resistance to heat transfer, it offers a relatively inexpensive method of bonding the copper tubing to the steel tank.

3.2.2. Evaluation of Condenser Parameters

A parametric study of the condenser design variables was performed in an effort to identify those parameters that influenced the design of the condenser the most (refer to Appendix II). The design variables and values used are presented in Table 3-3.

Table 3-3: Design Variables Used in Parametric Study

Parameter	Range
Type of Bond	Bare, Tape, Epoxy, Mastic, Solder
Temperature Difference	5°F, 10°F, 15°F, 25°F
Tube Diameter	¼ in. and 5/16 in.
Contact width	0.15, 0.1875, and 0.25 in.
Coil Spacing	1 in. and 2 in.

The required length, tube volume (refrigerant charge), and estimated pressure drop was calculated for numerous combinations of the parameters.

The results of the parametric study show that the largest contributor to the length of condenser tube was the type of bond and the temperature difference between the refrigerant and the water.

However, once the contact resistance (type of bond) was significantly reduced from that of a bare tank to that of a mastic, the additional gain of using solder was only slightly marginal.

We found that the tube diameter and contact width has little effect on the condenser tube length. However, the diameter of the tube has a significant impact on the pressure drop through the condenser. A sub-set of the initial results are presented in Table 3-4.

Table 3-4: Required Condenser Length for a Capacity of 3600 Btu/hr[4]

Bond	ΔT = 5°F	ΔT = 15°F	ΔT = 25°F
Bare Tube	438	134	77
Thermal Mastic	190	52	29
Solder	179	48	27

Upon completion of the contact resistance experiment, a similar analysis was performed using the contact resistance values obtained from the experimental work. A sub-set of the results are presented in Table 3-5.

Table 3-5: Required Condenser Length for a Capacity of 3600 Btu/hr

Bond	ΔT = 5°F	ΔT = 10°F	ΔT = 15°F
Bare Tube	498	236	153
Thermal Mastic	234	106	67
Solder	197	87	54

The design temperature difference of 10°F was selected. With a 10°F temperature difference, the condenser is 1) not extraordinarily long (as with 5°F), and 2) provides a reasonable upper limit to the maximum condensing temperature (at 135°F water temperature, the condensing temperature is not likely to exceed 145°F).

Finally, a cost analysis shows that the most cost-effective condenser is one that uses thermal mastic as the bond (see Table 3-6).

Table 3-6: Cost Comparison of Similar Condenser Designs

	Bare	Mastic	Solder Tape
Cost / Unit Length	$ 0.13	$ 0.172	$ 0.272
Length at 10°F	236	106	87
Total Cost	$30.68	$18.23	$23.66

Source: Manufacturer's and Vendor's data

3.2.3. Conclusion

The condenser should limit the maximum temperature difference between the water temperature and the refrigerant at 10°F. The most cost-effective condenser design is a 106 ft.-long, D-shaped, ¼ in., copper tube bonded to the water tank using thermal mastic.

3.3. Expansion Device Selection

The expansion device is one of the basic elements of a vapor compression heat-pump cycle. The basic function of the expansion device is to meter (or control) the refrigerant flow rate from the condenser into the evaporator, generally so that refrigerant flow into the evaporator is comparable to the mass flow rate of refrigerant vapor pumped by the compressor, and to the rate at which refrigerant boils off in the evaporator. The pressure drop across the expansion device separates the high-pressure side of the system from the low-pressure side.

Several types of expansion devices are common for use in heat pump and refrigeration cycles. In the context of the refinement of the design of the HPWH, the commonly used expansion devices were evaluated with respect to the design and performance criteria of the HPWH, and to find the best balance of low cost, reliability, and heat pump performance.

A trade study was performed on three expansion devices to determine which would be most suitable for the application of a HPWH:

1. A capillary tube,

2. An automatic expansion valve (also referred to as a constant-pressure expansion valve) (AXV), and

3. A constant superheat thermostatic expansion valve (TXV).

The capillary tube is the baseline expansion device to which both the AXV and TXV have been compared.

The following process was used to compare these three expansion devices:

*   The characteristics of each type of expansion device were identified;

*   Criteria for comparison were established;

*   The importance of each criterion was evaluated and weighing factors were established; and,

*   Each device was evaluated based on the criteria.

Section 3.3.1 describes the physical hardware configuration and the basic operating / control principles of each expansion device.

3.3.1. Description of Candidate Expansion Devices

As indicated above, the basic function of the expansion device is to control the refrigerant mass flow rate from the condenser to the evaporator so that it matches the refrigerant mass flow rate that is being pumped by the compressor. In the HPWH, this must occur over a wide range of condenser pressure corresponding to the tank water temperature falling anywhere between the cold water supply temperature and the hot water set point

temperature, and for evaporator pressures that vary with the ambient air temperature. As indicated in Table 3-7, the basic refrigerant mass flow rate response of a constant speed compressor and a fixed restriction (e.g. a capillary tube) flow control device to changes in the condenser or evaporator pressures are opposite to each other. In the brief descriptions that follow, the control response to offset this tendency is described for each expansion device.

Table 3-7: System Effects with Changes in Pressures

System Conditions	Effect On Compressor Mass Flow Rate	Effect On Flow Through A Fixed Restriction
Increased Condenser Pressure	Decrease	Increase
Increased Evaporator Pressure	Increase	Decrease

Capillary Tube

The capillary tube is a simple, fixed restriction metering device of a fixed length of small diameter copper tubing. Lengths typically are between 2 feet and 15 feet, and the typical inside diameter between $^1/_{32}$ and $^1/_8$ inch. Within the complete heat -ump system, a suction accumulator often accompanies the capillary tube to accommodate operating conditions where a large portion of the charge inventory is in the low-pressure side of the system.

Frictional resistance to flow through the capillary tube regulates the flow of refrigerant from the condenser to the evaporator. For a given capillary length and inside diameter, the refrigerant flow rate is controlled primarily by two factors — the high-to-low side pressure difference forcing the flow through the capillary, and the subcooling or vapor quality of the refrigerant entering the capillary. The flow is roughly proportional to the pressure difference, but increases significantly if the entering liquid is highly subcooled. Conversely, if the entering refrigerant is two-phase liquid/vapor mixture, the mass flow rate decreases significantly.

A capillary can provide refrigerant flow control (with the capillary mass flow rate being equal to the compressor mass flow rate) over a range of operating conditions. Balancing of capillary and compressor flow rates occurs as refrigerant charge transfer from high-side to low-side (or vice-versa) to decrease subcooling and increase vapor quality (or vice-versa), thereby decreasing (or increasing) the refrigerant flow rate.

The ASHRAE Refrigeration Handbook, Section 45, contains a more extensive functional explanation on the operating characteristics of capillaries and other expansion devices.

Automatic Expansion Device (AXV)

The automatic expansion valve (AXV), shown in Figure 3-5, is a mechanically adjusting metering device. It regulates the flow of refrigerant to maintain a constant pressure within the evaporator. It is composed of a needle valve, a spring with a tension adjusting screw, and a diaphragm. Within a refrigeration cycle, an accumulator may accompany the AXV.

The valve operates by using the balance of the forces between the evaporator pressure acting below the diaphragm and the spring pressure acting above it to position the valve needle. The difference between the two forces causes the diaphragm to move in the direction of the lower force, thereby drawing the needle valve into or out of its seat. If the evaporator pressure is higher than the predetermined spring pressure, the diaphragm will move upwards, closing the needle valve. This reduces the refrigerant flow into the evaporator thereby decreasing the pressure in the evaporator.
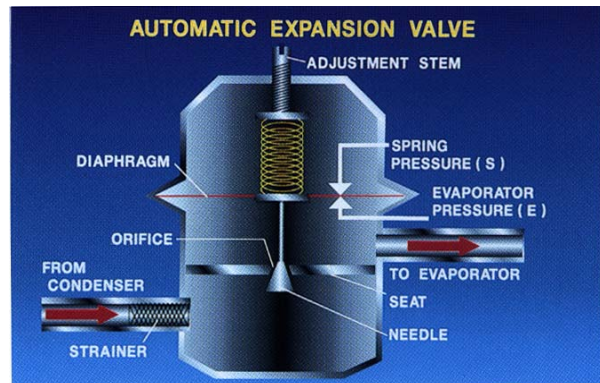
Figure 3-5: Automatic Expansion Valve

If the evaporator pressure is lower than the predetermined spring pressure, the diaphragm will move downwards, opening the needle valve. This increases the refrigerant flow into the evaporator which increases the pressure in the evaporator.

At high ambient temperatures, the AXV limits the flow of refrigerant into the evaporator to that which is consistent with the preset pressure set point. While this theoretically reduces the capacity and COP, it limits compressor motor power consumption, preventing overloads.

At low ambient temperatures, the AXV will be driven to the fully open position, in an attempt to reach the preset pressure. At this condition, refrigerant charge is transferred to the low-side (hence the potential need for a suction accumulator) and flow balance is achieved by virtue of uncondensed refrigerant vapor entering the valve. Thus, with an AXV, it is important not to oversize the valve and the condenser volume should be minimized to the extent practicable.

Thermostatic Expansion Device (TXV)

The thermostatic expansion valve (TXV), shown in Figure 3-6 is a mechanically adjusting metering device. It regulates the flow of refrigerant by maintaining a pre-defined amount of superheat leaving the evaporator, thereby ensuring that only vapor is leaving the evaporator, regardless of the operating conditions. It is composed of a needle valve, a spring with a tension adjusting screw, a diaphragm, and a remote sensing bulb that is clamped onto the outside wall of the suction line to the compressor. There may be a bleed port in the needle valve to assist in off-cycle system equalization. The sensing bulb is charged with the same refrigerant used as the heat-pump working fluid.

The valve operates to control the superheat in the line leaving the evaporator and going to the compressor. The valve operates by using the interaction of the forces between the sensing bulb pressure acting above the diaphragm and the spring and evaporator pressure acting below it. The difference between the two forces causes the diaphragm to move in the

direction of the lower pressure, thereby drawing the needle valve into or out of its seat. If there is too much superheat, the sensor bulb will increase the pressure on the diaphragm. This causes the needle valve to open, increasing the flow of refrigerant through the evaporator. If there is too little superheat, the sensor bulb will reduce the pressure on the diaphragm. This causes the needle valve to close, decreasing the refrigerant flow through the evaporator.
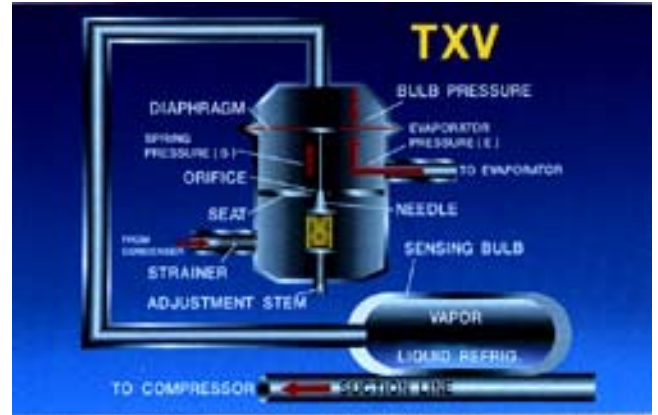
3.3.2. Definition of Criteria and Weighting Factors

Figure 3-6: Thermostatic Expansion Valve

The expansion devices were evaluated with respect to the criteria shown Table 3-8. A weighting factor was assigned to each criterion according to its importance to the design and success of the HPWH.

Table 3-8: Evaluating Criteria and Weighting Factors

Criteria	Weighting Factor
Applied Cost	5
System Reliability	5
System Efficiency	3
System Capacity	3
Behavior in Extreme Conditions	3
Off-Cycle Equalization	3
Valve Reliability	1
Ease of Manufacturing	1

A brief discussion of each criterion and the rationale for the assigned weighting follows below.

Applied Cost

As discussed throughout this report, a key objective of the development has been to minimize the cost of the HPWH in order to facilitate its acceptance in the market. Therefore, the cost of the expansion device was assigned the highest weight in the selection of the expansion device.

System Reliability

The system reliability is an important to the market acceptance of the HPWH. Based on prior market analysis, reliability is comparable to cost in importance, and, as such, has been weighted the same. Reliability of the HPWH is dependent on the reliable and operation of the compressor. The expansion valve serves as a device that meters the refrigerant through the evaporator and to the compressor. If the compressor receives too much liquid refrigerant, the reliability will be affected because liquid may enter the piston, thereby damaging the compressor. If the refrigerant entering the compressor is highly superheated or at too high a pressure, the entire compressor temperature may increase to the point where the compressor lubricant will break down, or the motor over-temperature switch will trip off, degrading the reliability and the performance of the compressor.

System Efficiency

The HPWH is an energy efficient alternative to electric and gas water heating. Therefore, due to the nature of the product, this is a criterion with moderate importance to the choice of the design of the expansion device and has been weighted accordingly.

System Capacity

If the expansion device cannot supply a high enough refrigerant flow rate to the evaporator over a portion of the operating range, the heat-pump capacity will be reduced, possibly resulting in more of the water heating load being met by the electric resistance back-up. For example, when the tank water is cold and the condensing pressure is low, a capillary may not pass enough refrigerant flow. A weighting factor of 3 (moderately important) has been assigned.

Behavior in Extreme Conditions

It is assumed the HPWH will be installed in one of three locations within a household: the closet, the basement, or the garage. In garage installations, the HPWH will be exposed to outdoor ambient air temperatures which exceed the flow or high temperature extremes of the design envelope. The HPWH will also be exposed to water temperatures at the extremes of the design envelope. For each combination of extreme ambient air and water temperature (e.g. high ambient air, low water temperature), the effect of the expansion device operating characteristics on the system reliability, the system efficiency, and the system capacity was evaluated. Performance at extreme conditions is a moderate concern, because the HPWH typically will be exposed the extreme ambient conditions for only a small portion of its life-cycle.

Off-Cycle Equalization

During normal operation of the HPWH, the compressor will cycle on and off as the tank water temperature falls below then exceeds the water temperature set point. During the off-cycle, it is important that the refrigerant pressures come to equilibrium in the system because of motor starting torque limitations. Off-cycles will tend to be long, so rapid equalization is not necessary.

Valve Reliability

The reliability of the heat-pump system depends on the reliability of the expansion device, because the system will not operate properly if the expansion device fails. All three of the candidate expansion devices have well established records for high reliability, so a low weighting factor was assigned.

Ease of Manufacturing

The ease of integrating the expansion device during manufacturing is a consideration when choosing an expansion device. The number of steps needed to integrate the device and its accessories into the refrigeration cycle adds to the manufactured cost of the HPWH. This cost, however, is not as significant as the cost of the device, so a lower weighting factor was assigned.

3.3.3. Evaluation of Each Expansion Device

Each of the expansion device options have been evaluated against the criterion described above. The capillary tube is the baseline against which the TXV and the AXV are compared.

Applied Cost

The applied cost of each expansion device is shown in Table 3-9. The capillary tube (with a small accumulator) is the least costly of the three options ($3 –$4). The AXV is the most expensive option, due to the need for an accumulator along with the valve.

Table 3-9: Cost Table for Expansion Device Options

Capillary Tube		Automatic Expansion Valve		Thermostatic Expansion Valve	
Equipment	Cost	Equipment	Cost	Equipment	Cost
Tube	$ 1	Valve	$11-$12	Valve/Bulb	$11-$14
Accumulator	$2-$3	Accumulator	$2-$3	N/A	N/A
Total	$3-$4		$13-$15		$11-$14

System Efficiency

At any given operating condition, the maximum level of heat-pump capacity and COP will be reached if the refrigerant flow control device meters a refrigerant flow rate that results in two phase (liquid / vapor) refrigerant flow through the entire evaporator, with no unevaporated liquid carried over from the evaporator outlet into the suction vapor line. If this condition is maintained, the heat transfer capacity of the evaporator is fully utilized. The vapor compressed by the compressor is only the vapor flashed through the expansion device plus the vapor that usefully absorbs heat in the evaporator. In addition, the high side refrigerant inventory can be maintained to provide an optimum level of subcooling. Commercially available expansion devices deviate from this ideal state, as described below.

Capillary Tube – A capillary is a fixed restriction, and as such will only provide the optimum restriction to match or approach the ideal situation for a narrow range of conditions. As described above, capillaries provide satisfactory refrigerant flow control over a wide range of conditions through the interaction of the high / low side charge distribution and the effect on subcooling / vapor content on the flow rate. At off-design conditions, several efficiency losses occur with capillaries:

- When the pressure difference is too large, the system adjusts by transferring charge to the low side of the system. Within a range, efficiency impacts are minimal, as the efficiency impact of the decrease in condensing pressure offsets the impact of decreased subcooling. When further charge transfer occurs, uncondensed vapor passes through the capillary and adds to the compressor load, without absorbing any heat. The transfer of charge to the low side also results in some liquid return to the compressor. While a small amount of liquid return helps to cool the compressor, it also adds to the compressor load when it vaporizes within the compressor shell.

- When the high to low pressure difference is to low to pass the necessary flow, charge transfers to the high side of the system, filling the condenser so that less condenser surface is available for condensing (and more is used for subcooling). The condensing pressure increases along with subcooling, gradually degrading the efficiency. As charge is removed from the low side, the evaporator becomes partially "starved" for refrigerant, which artificially drives the evaporator temperature and the pressure down, further reducing the efficiency.

Automatic Expansion Valve – **The AXV can deliver the ideal refrigerant flow at one ambient temperature level.** At the ambient temperature corresponding to the AXV pressure setting, the flow through the AXV will match the evaporator boiling rate, over the entire tank water temperature and pressure by "starving" the evaporator, reducing the efficiency from the potential?, given the higher ambient temperature. At lower ambient temperatures, the AXV will be fully open in an attempt to reach the set point pressure, and flow balance will occur via vapor entering the valve from the condenser, as refrigerant inventory shifts to the low side of the system. Efficiency is lost due to both the energy necessary to recompress the uncondensed vapor and due to liquid carryover to the compressor.

Thermostatic Expansion Valve – With a TXV, ideal refrigerant flow control is approached. The requirement for superheat leaving the evaporator (to allow the valve to control) results in less than full utilization of the evaporator heat transfer capacity and a depression of the evaporator pressure / temperature by a small amount. With a low superheat setting, the impact on the COP and capacity are small.

System Reliability

Capillary Tube – **The capillary tube will help prevent the compressor from overheating.** At high condensing pressure, it allows some liquid refrigerant to travel back to the compressor, where it will vaporize and cool the compressor. However, this can also be a negative attribute if too much liquid gets into the compressor and damages the pistons; the purpose of the accumulator is to limit the rate of liquid refrigerant flow to the compressor.

Automatic Expansion Valve – The AXV protects the compressor by limiting the inlet pressure to the level recommended by the compressor manufacturer, preventing motor overload. At high tank temperature and normal and high ambients, the refrigerant vapor will be superheated at the compressor inlet, and the compressor discharge temperature will be at the high end of the recommended range. At lower ambient temperatures, liquid carryover from the suction accumulator back to the compressor will provide additional cooling.

Thermostatic Expansion Valve – **The TXV will prevent liquid refrigerant from slugging into the compressor.** The function of a TXV is to provide a constant amount of superheat, preventing unevaporated liquid from leaving the evaporator and slugging back to the compressor. However, the TXV does not allow any liquid refrigerant back to the compressor for cooling purposes. In hotter ambient conditions, this may lead to an overheated compressor and increased evaporator pressure.

System Capacity

Capillary Tube – **The capillary tube will have a comparatively low capacity at two conditions.** When the tank water is cold and the condenser pressure is correspondingly low, the refrigerant flow rate may be insufficient, thereby reducing capacity. At high water temperature and correspondingly high condensing pressure, some uncondensed refrigerant vapor will enter the capillary and some liquid refrigerant will be allowed to travel back to the compressor. This is a positive attribute because the condensing temperature will be as low as possible (given the tank water temperature) and the liquid refrigerant will vaporize and cool the compressor. However, a portion of the compressor's capacity is used to compress the refrigerant that evaporated in the compressor, thereby reducing the flow of refrigerant vaporized in the evaporator where heat is absorbed from ambient. The impact on capacity in both of these operating regimes can be minimized by judiciously minimizing the condenser volume and using a suction accumulator to limit liquid carry-over back to the compressor.

Automatic Expansion Valve – **The AXV will decrease capacity by reducing the refrigerant flow rate through the compressor.** In order to maintain the preset pressure in the evaporator, the AXV acts in either of two modes. At high ambient temperatures, the AXV decreases the flow through the valve to reduce a high pressure in the evaporator, thereby decreasing the capacity of the system. If the evaporator pressure is below the preset pressure, the AXV will fully open, increasing the flow through the valve. With the AXV fully opened, the refrigerant inventory shifts from the high side to the low side. On the high side, the refrigerant inventory falls so that a liquid-vapor mixture enters the AXV. This reduces the mass flow through the AXV to match the compressor mass flow rate. A suction accumulator is needed after the evaporator to hold the extra charge transferred from high side of the system. While this arrangement maintains the evaporating pressure at the highest level possible for the low ambient temperature, capacity is lost due to the uncondensed vapor that passes through the AXV and due to the vaporization of the small amount of liquid refrigerant that is returned to the compressor.

Thermostatic Expansion Valve – **The TXV will maintain near maximum capacity by providing moderately superheated refrigerant to the compressor.** By maintaining the preset level of superheat to the compressor, the TXV maintains the optimum refrigerant

flow through the evaporator. Achieving and maintaining a certain amount of superheat indicates that the compressor is only expending energy on the compression of refrigerant vapor that gained heat from the ambient air around the HPWH. Therefore, the TXV will have a comparatively high capacity.

Behavior in Extreme Conditions

The system reliability, system efficiency, and system capacity has been evaluated for each of the four combinations of ambient temperature extreme (~ 35°F and 95°F) and tank water temperature extreme (~35°F and 135°F). Each was rated by its similarity to (or deviation from) the operation in normal or typical operating conditions.

If the system behavior in the extreme conditions was similar to its behavior in the typical conditions, it was assigned a rating of "0". If the system behavior in the extreme conditions was better that its behavior in the typical conditions, it was assigned a rating of "+". If the system behavior in the extreme conditions was worse than its behavior in the typical conditions, it was assigned a rating of "-".

Overall, with a thermostatic expansion valve, the system will perform similarly at the four extremes to performance during normal conditions. It was chosen as the expansion valve that performed the best in extreme conditions. The results of the exercise gave the capillary tube a rating of –4, the automatic expansion device a rating of –10, and the thermostatic expansion valve a 6. The following explains and details the ratings given each expansion device.

Capillary Tube

Table 3-10 summarizes our ratings of capillary behavior at extreme conditions.

Table 3-10: Behavior of the Capillary Tube in Extreme Conditions

Extreme Condition	System Reliability	System Efficiency	System Capacity
High Ambient Temperature / Low Tank Temperature	0	-	-
High Ambient Temperature / High Tank Temperature	0	0	0
Low Ambient Temperature / Low Tank Temperature	0	0	0
Low Ambient Temperature / High Tank Temperature	0	-	-
Total = - 4	0	-2	-2

High Ambient Temperature / Low Tank Temperature

In these conditions, the pressure differential between the high and low sides forces insufficient refrigerant flow, thereby depressing the evaporator pressure, which then results in a low compressor flow rate.

System Reliability –While high superheat could occur at this condition, over-temperature will not occur due to the low compressor discharge pressure and the low load on the compressor motor.

System Efficiency – The low pressure in the evaporator decreases the efficiency of the refrigeration cycle.

System Capacity –There is a lower capacity due to a low flow rate. As discussed above with a low volume condenser, liquid backing up into the condenser will increase the high side pressure and liquid subcooling, increasing the flow and minimizing the capacity loss.

High Ambient Temperature / High Tank Temperature

In these conditions, the pressure differential suffices (typical of normal operating conditions) to pass refrigerant through the fixed restriction. There is a pressure differential typical of normal operating conditions between the high side and the low side of the system. The compressor inlet vapor may have some superheat.

System Reliability – Some superheat may appear at the inlet to the compressor, but not enough to overheat the compressor. This will have little effect on the reliability of the system.

System Efficiency – Due to the normally occurring differential pressure in the capillary tube and a small amount of superheat, the efficiency will not differ from the efficiency during typical operating conditions.

System Capacity – Due to the normally occurring differential pressure in the capillary tube, the capacity will not differ from the capacity during typical operating conditions.

Low Ambient Temperature / Low Tank Temperature

In these conditions, there is less pressure differential than at normal operating conditions but the reduced flow will (roughly) match the compressor flow rate at the reduced evaporator temperature. The capillary tube may pass too much liquid from the evaporator; however, a system with a properly sized accumulator will prevent the liquid from entering the compressor.

System Reliability – A properly sized accumulator will prevent liquid from entering the compressor. Therefore, the capillary tube will have little effect on the reliability of the system. The load on the compressor motor is minimal, so no cooling is needed.

System Efficiency – Due to the normally occurring differential pressure in the capillary tube and a small amount of superheat, the efficiency will not differ from the efficiency during typical operating conditions.

System Capacity – Due to the normally occurring differential pressure in the capillary tube, the capacity will not differ from the capacity during typical operating conditions.

Low Ambient Temperature / High Tank Temperature

In these conditions, there is a large enough pressure differential to pass a high flow rate of refrigerant through the fixed restriction. The high flow rate will result in liquid carryover from the evaporator; however, a system with a properly sized accumulator will prevent excessive liquid from entering the compressor.

System Reliability – A properly sized accumulator will prevent excessive liquid from entering the compressor. The small amount of liquid carried over from the accumulator to the compressor provides needed cooling. Therefore, the capillary tube will have little effect on the reliability of the system.

System Efficiency – The system efficiency will decrease moderately due to the use of a portion of the compressor capacity to compress uncondensed vapor passing through the capillary (a few percent of the total mass flow) and the vapor from the liquid refrigerant returned to the compressor (also a few percent of the total mass flow, which provides useful compressor cooling).

System Capacity – There will be a moderate decrease in the capacity of the refrigeration cycle, due to the uncondensed vapor passing through the capillary and the evaporation of liquid refrigerant in the compressor.

Automatic Expansion Valve

Table 3-11 summarizes our ratings of AXV behavior at extreme conditions.

Table 3-11: Behavior of the Automatic Expansion Valve in Extreme Conditions

Extreme Condition	System Reliability	System Efficiency	System Capacity
High Ambient Temperature / Low Tank Temperature	-	-	-
High Ambient Temperature / High Tank Temperature	-	-	-
Low Ambient Temperature / Low Tank Temperature	0	-	-
Low Ambient Temperature / High Tank Temperature	0	-	-
Total = -10	-2	-4	-4

High Ambient Temperature / Low Tank Temperature

In a high temperature ambient environment, there would be high pressure in the evaporator if sufficient refrigerant were supplied. The AXV will allow less refrigerant to flow through the evaporator, thereby artificially lowering the evaporator pressure. The lowered flow rate and evaporating temperature may be accompanied by a high level of superheat back to the compressor.

System Reliability – Due the low flow rate, the excessive superheat may overheat the compressor, thereby diminishing the system's reliability. At the low head pressure involved, this is a minor issue.

System Efficiency – Due to the artificially low pressure in the evaporator, there will be a decrease in the efficiency of the refrigeration cycle.

System Capacity – Due to the artificially low pressure in the evaporator, there will be a decrease in the capacity of the refrigeration cycle.

High Ambient Temperature / High Tank Temperature

In a high temperature ambient environment, there would be high pressure in the evaporator if sufficient flow of refrigerant were supplied to it. The AXV will allow less refrigerant to flow through the evaporator, thereby artificially lowering the evaporator pressure. The low flow rate may allow excessive superheat back to the compressor.

System Reliability – Due the low flow rate and high head pressure, the excessive superheat may overheat the compressor, thereby diminishing the system's reliability. By limiting the evaporator pressure to a level consistent with the operating range recommended by the compressor manufacturer, motor overloading is prevented and discharge temperature level could be kept within the recommended range.

System Efficiency – Due to the artificially low pressure in the evaporator, there will be a decrease in the efficiency of the refrigeration cycle.

System Capacity – Due to the artificially low pressure in the evaporator, there will be a decrease in the capacity of the refrigeration cycle.

Low Ambient Temperature / Low Tank Temperature

In a low temperature ambient environment, there is low pressure in the evaporator. The AXV allows more refrigerant to flow through the evaporator in order to increase the pressure. This may cause flooding of the evaporator coil and allow liquid refrigerant to leave the evaporator. However, a system with a properly sized accumulator should prevent the liquid from entering the compressor.

System Reliability – A properly sized accumulator will prevent liquid from entering the compressor. Therefore, the AXV will have little effect on the reliability of the system.

System Efficiency – More than enough refrigerant flow is allowed, allowing the evaporating temperature to be as high as possible relative to ambient conditions. However, uncondensed vapor will pass through the AXV at this condition, thereby reducing the efficiency by a few percent.

System Capacity – Capacity will be impacted similarly to the efficiency impact.

Low Ambient Temperature / High Tank Temperature

In a low temperature ambient environment, there is low pressure in the evaporator. The AXV allows more refrigerant to flow through the evaporator in order to increase the pressure. This may cause flooding of the evaporator coil and allow liquid refrigerant to leave the evaporator. A system with a properly sized accumulator should prevent the liquid from entering the compressor.

System Reliability – A properly sized accumulator will prevent liquid from entering the compressor. Therefore the AXV will have little effect on the reliability of the system.

System Efficiency – Liquid carry-over from the accumulator cools the compressor. The AXV will be fully open at this condition in an attempt to raise the evaporator pressure above the level attainable with low temperature ambient air as the heat source. As a result, the heat transfer capacity of the evaporator will be fully utilized. Due to the high condensing pressure, however, a significant amount of uncondensed vapor will pass through the AXV, reducing system efficiency and capacity. Unevaporated liquid carried over from the evaporator will be stored in an accumulator, and the small amount of liquid carried over from the accumulator will provide needed compressor cooling.

System Capacity – Capacity will be impacted similarly to the efficiency.

Thermostatic Expansion Valve

Table 3-12 summarizes our ratings of TXV behavior at extreme conditions.

Table 3-12: Behavior of the Thermostatic Expansion Valve in Extreme Conditions

Extreme Condition	System Reliability	System Efficiency	System Capacity
High Ambient Temperature / Low Tank Temperature	0	+	+
High Ambient Temperature / High Tank Temperature	-	+	+
Low Ambient Temperature / Low Tank Temperature	0	+	+
Low Ambient Temperature / High Tank Temperature	-	+	+
Total = 6	-2	4	4

High Ambient Temperature / Low Tank Temperature

In these conditions, the TXV will maintain a constant amount of superheat back to the compressor. High pressure in the evaporator will be maintained, corresponding to the high ambient air temperature, and refrigerant flow rate through the evaporator and compressor will increase.

System Reliability – Due to a high evaporating pressure, the compressor motor load will be high. While the superheated suction vapor will not provide extra cooling, (given the low condensing temperature), the compressor is not likely to overheat.

System Efficiency –The increase in refrigerant flow rate and high evaporating pressure will increase the efficiency of the system.

System Capacity – The increase in refrigerant flow rate and high evaporating pressure will increase the capacity of the system.

High Ambient Temperature / High Tank Temperature

In these conditions, the TXV will maintain a constant amount of superheat back to the compressor. A high pressure in the evaporator will be maintained and the refrigerant flow rate through the evaporator and compress will increase.

System Reliability – Due to a high evaporating pressure and a high condenser pressure, the compressor motor load is at a maximum. With superheated return vapor, overheating the compressor is a possibility. This will decrease the reliability of the system.

System Efficiency – The TXV will supply enough refrigerant flow to raise the evaporator pressure to a level corresponding to the high ambient temperature, thereby increasing the efficiency.

System Capacity – Due to the increase in the refrigerant flow rate through the evaporator, there will be an increase in the capacity of the refrigeration cycle.

Low Ambient Temperature / Low Tank Temperature

In these conditions, the TXV will maintain a constant amount of superheat back to the compressor. The evaporator pressure will correspond to the low ambient temperature without any excess liquid or uncondensed vapor flow, obtaining near maximum efficiency for this set of conditions.

System Reliability – The TXV will maintain a constant amount of superheat back to the compressor. At this low motor load conditioning, overheating will not occur. Hence, the system reliability will be similar to that during normal operating conditions.

System Efficiency – As described above, near optimum efficiency will be obtained (for this set of conditions).

System Capacity – As described above, near optimum capacity will occur (for this set of conditions).

Low Ambient Temperature / High Tank Temperature

In these conditions, the TXV will maintain a constant amount of superheat back to the compressor. A low pressure in the evaporator will be maintained with a corresponding decrease in the refrigerant flow rate through the evaporator.

System Reliability – The TXV will maintain superheat back to the compressor. At the low evaporating pressure, the motor load will not be excessive, but at the high lift, the compressor discharge temperature will be near recommended limits. Long-term reliability might be negatively affected, compared to that realized with the other expansion devices which return a moderate amount of liquid refrigerant to the compressor in this condition.

System Efficiency – The evaporator temperature will approach the maximum level possible at the low ambient temperature, without excess liquid or vapor flow. As a result, system efficiency and capacity will approach the maximum possible for this set of conditions.

System Capacity – System capacity will approach the maximum possible at this set of conditions.

Off-Cycle Equalization

As discussed previously, the starting torque required for a small, single cylinder reciprocating refrigerant compressor can exceed the available standing torque of the compressor motor, unless the high and low side pressure have substantially equalized during the off-cycle. For this to occur at the maximum tank temperature, (which indicates the off-cycle), it is necessary for the majority of the refrigerant charge to pass through the expansion device to the low side of the system. The off-cycle is usually long (hours), so a rapid transfer of refrigerant is not necessary.

Capillary Tube – **The capillary tube will allow refrigerant equalization during off-cycle.** During off-cycle, the fixed restriction of the capillary tube remains open. This will provide equalization of high and low pressure during off-cycle.

Automatic Expansion Valve – **The AXV will not allow full equalization during off-cycle if the ambient air temperature is higher than the refrigerant saturation temperature at the AXV pressure setting.** The AXV will admit refrigerant into the evaporator until the pressure exceeds the AXV setting. When the compressor is off, very little refrigerant needs to be added to the evaporator for the evaporator pressure to reach the AXV setting, unless the ambient temperature is very low (which will not occur in indoor installations).

Thermostatic Expansion Valve – **The TXV with a bleed port will allow refrigerant equalization during off-cycle.** During off-cycle, the TXV allows for refrigerant to flow through a bleed port integrated into the needle valve. This will provide a slow equalization of high and low side pressures during the off-cycle.

Valve Reliability

Capillary Tube – **The capillary tube is very reliable.** A capillary tube is a piece of copper tubing that provides a pressure drop due to its frictional resistance. As there is no moving parts, the failure rate is extremely low. Therefore, it is extremely reliable. Failure can only occur if the device is clogged by contamination in the system. System cleanliness and exclusion of moisture at assembly are the keys to avoiding this failure.

Automatic Expansion Valve – **The AXV is highly reliable; however, there is a risk of failure due to moving parts.** An AXV is essentially a needle valve. Due to the fact that

the AXV is a proven technology, it is highly unlikely that there will be failure within the valve due to the moving parts. The only other means of failure is the possibility that a portion of the valve may become clogged by contamination in the system.

Thermostatic Expansion Valve – **The TXV is highly reliable; however, there is a risk of failure due to moving parts.** A TXV is essentially a needle valve. Due to the fact that the TXV is a proven technology, it is highly unlikely that there will be failure within the valve due to the moving parts. The only other means of failure is that a portion of the valve may become clogged by a piece of debris the possibility in the system.

Ease of Manufacturing

Capillary Tube – **The integration of the capillary tube and accumulator consists of four brazed joints.** During the manufacturing of the HPWH, the capillary tube is attached by brazing one end of the capillary tube to the exit of the condenser and brazing one end to the inlet of the evaporator. However, a capillary tube should also have an accumulator. To integrate the accumulator into the refrigeration cycle, it must be attached in the suction line of the compressor. The accumulator should be attached in series with the suction line, having both ends being brazed. Therefore, there are four brazed joints to install the capillary tube and the accumulator.

Automatic Expansion Valve – **The integration of the AXV and accumulator will consist of four brazed joints.** During the manufacturing of the HPWH, the AXV is attached by brazing one end of the AXV to the exit of the condenser and brazing one end to the inlet of the evaporator. However, the AXV also needs an accumulator. To integrate this into the refrigeration cycle, it must be attached in the suction line of the compressor. The accumulator should be attached in series with the suction line, having both ends brazed. Therefore, there will be four brazed joints to install the AXV and the accumulator.

Thermostatic Expansion Valve – **The integration of the TXV consists of two brazed joints, a copper strap, and insulation of the sensor bulb.** During the manufacturing of the HPWH, the TXV is attached by brazing one end of the TXV to the exit of the condenser and brazing one end to the inlet of the evaporator. The sensing bulb is attached to the suction line of the compressor using a copper strap and then insulated.

3.3.4. Results and Conclusions

Using the criteria from the previous sections, the TXV was determined to be the most suitable expansion device for the HPWH application. Table 3-13 compiles the results of the evaluations described above.

If the expansion device candidate's behavior is similar to that of the capillary tube, the baseline, it was assigned a rating of "0". If the expansion device candidate's behavior is much worse than that of the capillary tube, the baseline, it was assigned a rating of "-3". If the expansion device candidate's behavior is much better than that of the capillary tube, the baseline, it was assigned a rating of "+3".

Table 3-13: Expansion Device Design Matrix

Criteria	Weight	Capillary Tube	Automatic Expansion Valve		Thermostatic Expansion Valve	
Cost	5	0	-3	**-15**	-3	**-15**
System Reliability	5	0	+1	**+5**	-1	**-5**
System Efficiency	3	0	+1	**+3**	+1	**+3**
System Capacity	3	0	+2	**+6**	+2	**+6**
Behavior in Extreme Conditions	3	0	-2	**-6**	+3	**+9**
Off-Cycle Equalization	3	0	-3	**-9**	0	**0**
Valve Reliability	1	0	-1	**-1**	-1	**-1**
Integration with HPWH	1	0	0	**0**	-2	**-2**
	Total	0	**-17**		**-5**	

3.4. Condensate Management System (CMS)

In order for the HPWH to be considered a "drop in" water heater, it must be capable of disposing condensate without access to a condensate drain. The subsections below describe the following considerations:

- The functional requirements of the Condensate Management System, evaluation of various evaporation methods, and comparison of viable design concepts. The functional alternatives, the criteria used to compare the different options, and the results of the Functional Evaluation are also described;

- Description of the positives and negatives for each design detail component;

- Description of the conceptual design; and

- Conclusion and recommendations.

While we evaluated the CMS design, in concurrence with the Commission Contract Manager, we did not incorporate a CMS into the prototype design or the field test.

3.4.1. Functional Requirements

The objective of the Condensate Management System (CMS) is to meet the following functional requirements:

- The CMS must evaporate condensate into the surrounding air;

- The CMS must be capable of evaporating condensate when the heat pump is off-cycle;

- The CMS must be designed to prevent condensate overflow;

- The CMS must be able to store a small amount of condensate while the heat-pump is operating;

- The CMS must incorporate an optional condensate drain feature;

- The CMS must be designed to handle worst-case (near saturated) conditions; and

- The CMS cannot run when either of the back-up resistance elements are on because of amperage limitations.

3.4.2. Assumptions

Along with the functional requirements, certain assumptions shaped the conceptual design of the CMS:

- Though a condensate drain is the preferred and most simple design, it is not a viable option for all installations;

- The CMS must evaporate condensate at the same rate it's formed;

- The HPWH's current envelope cannot be increased; and

- Laboratory testing of the second-generation prototype established the maximum condensation formation rate as 1 lbm/hr at 80 $^{\circ}$F 80% RH.

3.4.3. Design Process

The conceptual design process for the CMS consisted of three stages: Functional Evaluation, Detail Evaluation, and Conceptual Design, with the latter two stages being dependent on the results of the first stage.

During the Functional Evaluation, various re-evaporating methods were compared in the context of design criteria. The second stage of the design entailed considering the design details, including the location of the condensate container, associated with the previously selected functional method. The optimum combination of these details resulted in the Conceptual Design.

3.4.4. Functional Evaluation

The two methods were considered in the form of four functional designs:

1. Electrical Resistance;

2. Desuperheater;

3. Integrated with Tank; and

4. Integrated with Compressor.

Table 3-14, Figure 3-7, and Figure 3-8 describe the details of the functional alternatives.

Table 3-14: Functional Alternatives

Functional Alternative	Type	Description
Electrical Resistance	Evaporation by Boiling	Uses an electric element to boil the condensate.
Desuperheater	Sub-boiling Evaporation	Uses the hot discharge gas from the compressor to evaporate the condensate.
Integrated with Tank	Evaporation	Uses condenser heat to evaporate the condensate.
Integrated with Compressor	Evaporation	Uses waste compressor heat to evaporate condensate.
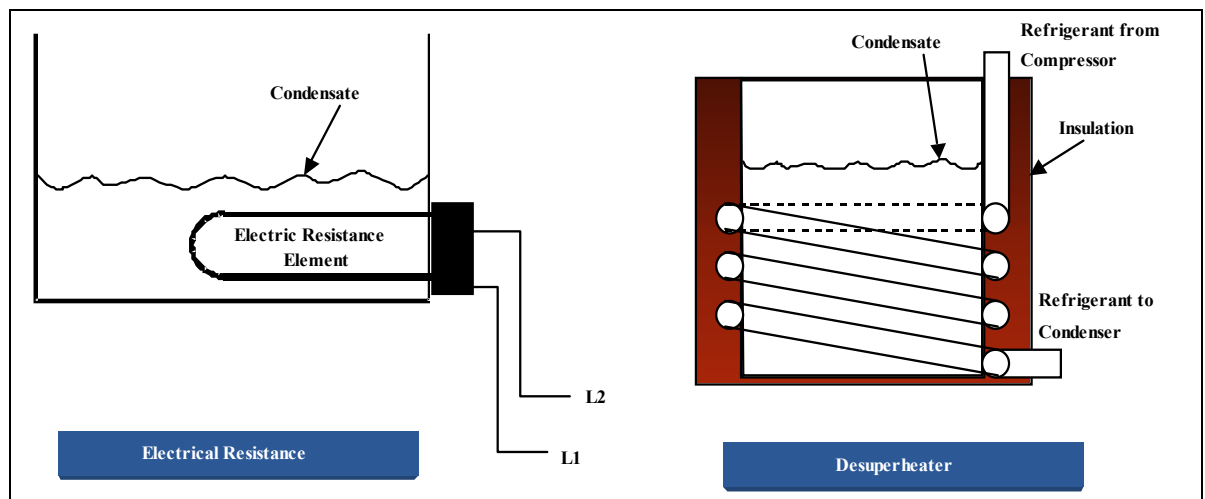
Figure 3-7: Functional Alternatives for the CMS
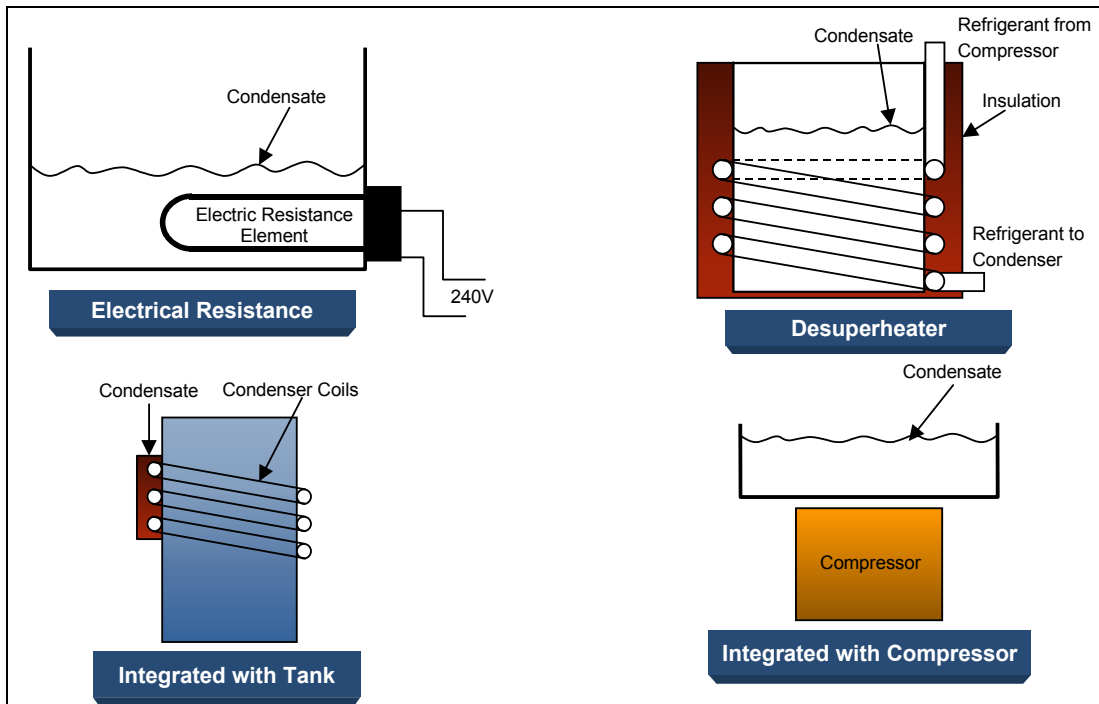
Figure 3-8: Additional Functional Alternatives for the CMS

3.4.5. Definition of Criteria and Weighting Factors

The functional alternatives were compared relative to criteria, which are listed in Table 3-15. Consideration of the functional requirements, the impact of the CMS on the overall system, and the off-cycle performance of the CMS led to the criteria. We assigned weighting factors to each criterion to reflect its relative importance. High importance was given to Worst Case Performance, Performance Impact, Manufacturability, and Reliability. These criteria define the functionality of the CMS, while Off-Cycle Evaporation merely characterizes performance.

Table 3-15: Functional Alternative Design

Criteria	Weighting Factor	Description
Worst-Case Performance	5	Ability to evaporate condensate during hot, humid conditions.
Performance Impact	5	Effect on Overall Delivery Efficiency (ODE).
Manufacturability	5	Feasibility, ability to be integrated with the HPWH, and relative manufacturing cost.
Reliability	5	Ability to evaporate condensate repeatedly and consistently.
Off-cycle Evaporation	3	Ability to remove condensate when the heat pump is off.

3.4.6. Result of Functional Evaluation

As seen in Table 3-16, Electric Resistance compares favorably to the other functional types. For comparison purposes, the Electric Resistance option was used as the baseline. Each of the three other options was compared to the baseline for each criterion. The functionalities were given a positive point, no points, or a negative point depending on how it compared to compared to the Electric Resistance option.

Table 3-16: Functional Comparison

Criteria	Weighting Factor	Electric Resistance	De-Superheater	Integrated with Tank	Integrated with Compressor
Worst-Case Performance	5	0	-	-	-
Performance Impact	5	0	0	0	+
Manufacturability	5	0	0	0	0
Reliability	5	0	-	-	-
Off-cycle Evaporation	3	0	-	-	-
*	TOTAL	0	-13	-13	-8

Worst-Case Performance

Sub-boiling-point evaporation becomes more difficult as the surrounding air approaches saturation. Boiling, on the other hand, can evaporate condensate into highly saturated air. The Electric Resistance option is the only alternative that boils the condensate, so it performs better at near-saturated conditions.

Performance Impact

The Electric Resistance option evaporates condensate at an efficiency of about one. The three remaining options, in one way or another, utilize heat generated at heat-pump efficiency (3 or 4) to evaporate condensate and, hence, are inherently more energy efficient than Electric Resistance.

For comparison purposes, we estimated the energy impacts at a condensate formation of 0.5 lbm/hr (half the maximum condensate formation rate) (Table 3-17). The energy impacts are reflected in the Overall Delivery Efficiency, which is a measure of a water heater's efficiency under actual operating conditions.

Table 3-17: ODE Comparison

	Electrical Resistance	Desuperheater
Nominal HPWH Power Input (Btu/h)	2066.5	1500
Nominal Heating Capacity (Btu/h)	3000	2463
Energy for CMS (Btu/h)	566.5	537
Overall Delivery Efficiency (ODE)	1.45	1.64

The energy savings advantage of the Desuperheater compared to the Electrical Resistance option is approximately 12%, which does not constitute a definitive advantage. On average, the ODE of the HPWH is not dependent on the CMS selection.

Manufacturability

Each functional alternative has similar manufacturing impacts.

Reliability

The ability to evaporate condensate consistently depends on the amount of heat associated from each functional alternative. Since the Electric Resistance alternative can produce boiling temperatures, it can evaporate condensate at near-saturated ambient conditions. The other alternatives, however, rely on sub-boiling evaporation (compressor discharge temperatures are generally under the normal boiling point of water). Therefore, the Electric Resistance alternative can dispose of condensate more reliably.

Off-Cycle Evaporation

Off-cycle evaporation requires independence from the heat-pump for energy input. The Electric Resistance Option is independent of the heat-pump system. Off-cycle evaporation for the Desuperheater and Integrated with Compressor options entails natural convection without heat input. The Integrated with Tank alternative provides heat when the heat-pump is off, but drawing heat from the tank provides the lowest-temperature heat (compared to the other alternatives) and, hence, the slowest evaporation rate. This increases the chances that not all condensate will be evaporated before the next heat-pump cycle.

3.4.7. Detailed Evaluation

Detailed Evaluation, the second stage of the CMS design, depends on the choice of the functional alternative. The details consequently rely on using a resistive element as part of the final conceptual design. This section evaluates options for the various design components: Location, Heater Selection, Sensor Type, and Container Size.

Location

The container for the CMS can exist either inside the shroud or outside the shroud. Various options exist for each location. For example, inside the shroud, the container can merely be the condensate drain pan or be separate from the drain pan. The separate container can be

positioned above or below the drain pan, yet either option requires increased cost, especially if a pump is required for upward flow, and an increased evaporator assembly height.

The most logical placement of the container outside the shroud is along the tank's side. The container must be placed in the insulation layer around the tank so as not to increase the HPWH envelope.

Heater Selection

Preferred heater type and heater capacity must be determined before selection of the heater.

Heater Type

The electric resistance heater can be either immersed in the condensate or placed on the outer surface of the condensate container. Table 3-18 presents the advantages and disadvantages associated with each option.

Table 3-18: Heater Type Comparison

Type	Advantages	Disadvantages
Immersion	• Direct contact with water minimizes heat losses • Higher capacity capabilities	• Potential for burn-out due to higher watt density
Surface	• Less likely to overheat due to lower watt density	• Must be formed to surface, may need adhesive

Heater Capacity

The heating capacity is dependent on the required evaporation rate. Under the assumptions that on-cycle evaporation is necessary and that condensate is not actively stored, the evaporation rate must equal the maximum condensation rate. When the relative humidity exceeds 70%, the HPWH can potentially generate upwards of one pound of condensate per hour.[2] Thus, the heating capacity must be capable of evaporating one pound of condensate an hour. The heat required to bring the liquid condensate from a temperature of 50 ^{o}F to a vapor state of approximately 70 ^{o}F is 1150 Btu/h, which translates to a heater capacity of 340 watts. While the ambient air temperature may exceed 70 ^{o}F, the heater capacity required is relatively insensitive to air temperature.

3.4.8. Sensor Type

Table 3-19 identifies four sensor alternatives. Because of cost advantage, the float switch and the diaphragm pressure switch are favorable. The diaphragm pressure switch, typically included in washing machines, dishwashers, and water treatment systems, provides multi-level control capability, which allows for the start, stop, and overfill-prevention functions to be satisfied with one sensor. The drawback to using the pressure switches is that the diaphragms may be susceptible to failure if exposed to liquid and the air trap must periodically be vented to atmosphere.

While some float switches are capable of the three control functions, multiple moving floats are required resulting in higher failure rates. Additional floats on one assembly require a specific container width and depth. The alternative to having multiple floats on one assembly is to have two separate switches.

Table 3-19: Sensor Alternatives

Sensor Type	Cost	Description
Float Switch	$10 and up	Uses a float to sense level.
Optic Switch	$100 and up	Contains an infrared LED and light receiver. Light from the LED is directed into a prism in the tip of the sensor. As fluid covers the prism, the light is no longer detected.
Conductive Switch	$65 and up	Utilizes the conductivity of the liquid itself. As the cut-to-length electrodes come into contact with the fluid or conversely are exposed, the switch opens and closes.
Diaphragm Pressure Switch	$10 and up	Operates on change in pressure due to the weight of liquid acting on an air chamber. Trip points are set at specific pressures.

3.4.9. Results of Detailed Evaluation

Two combinations of design components, listed in Table 3-20, represent viable alternatives for the Condensate Management System. The alternatives differ in two ways: location and sensor type. The Along the Tank's Side design takes advantage of the available water column pressure to use a pressure diaphragm switch, while the Single Drain Pan design must use float switches (since insufficient water column is available to operate a pressure switch). Both combinations include the same capacities and heater types. The surface heater is preferred over the immersion style because the surface type is less susceptible to burn out.

Table 3-20: Component Combinations

Component	Design 1	Design 2
Location	Single Drain Pan	Along Tank's Side
Heater Type	Surface	Surface
Sensor Type	Float Switches	Pressure Diaphragm
Heater Capacity (Btu/h)	1150	1150
Container Capacity (lbs.)	1	1

3.4.10. Conceptual Design

The following compares two concepts (which integrate the component combinations listed above) by evaluating the designs relative to criteria of unequal weight.

Description of Design Concepts

The Single-Drain-Pan Concept, illustrated in Figure 3-9, incorporates a drain pan having two sections: a slanted section and a heated section. The slanted section guides the condensate through perforations into the heated section to be evaporated. The heated section extends from the evaporator assembly so that the re-evaporated condensate can be blown out of the shroud by the evaporator fan. The perforated divider reduces air leakage around the evaporator. The heating element (not shown in the figure) is attached to the bottom of the heated section, while two float switches are used to accomplish the three control functions. Two switches are required to ensure both storage and overfill prevention. Using only one sensor would sacrifice one of the functions.
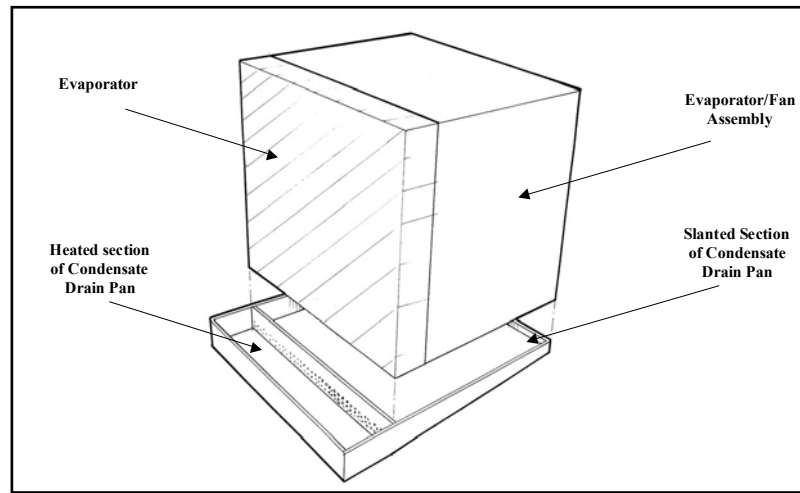
Figure 3-9: The Single-Drain-Pan Concept

The Along-the-Side Concept (see Figure 3-10) consists of a heating element wrapped around the cylindrical container and a diaphragm pressure switch. The multi-function capability of the pressure switch allows the heat-pump to remain on during condensate re-evaporation. The container is mounted in the insulation around the tank.
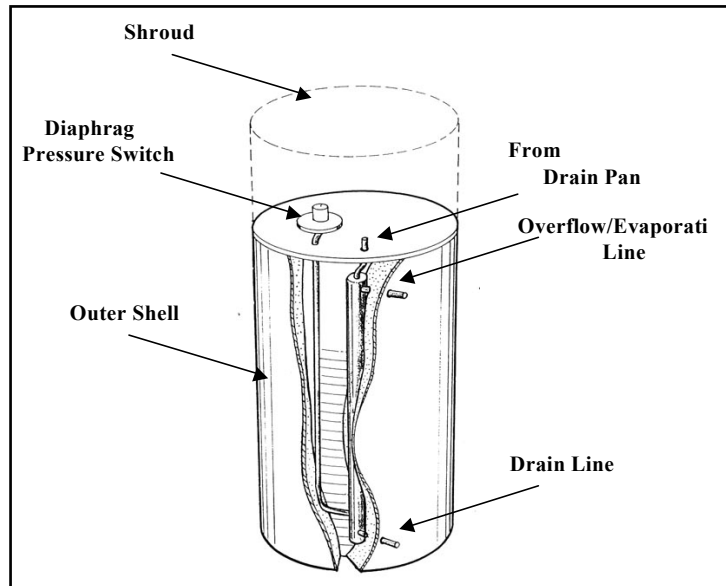
Figure 3-10: Along-the-Side Concept

Definition of Criteria and Weighting Factors

The two concepts were compared relative to criteria, which are listed in Table 3-21. Similar to the evaluation of the functional alternatives, consideration of the functional requirements, the impact of the CMS on the overall system, and the off-cycle performance of the CMS led to the criteria. High importance was given to the criteria which characterized the integration with the HPWH, serviceability, and the performance and control capabilities of the CMS. The three criteria of lesser importance characterize the physical aspects of the design.

Table 3-21: Conceptual Design Criteria

Criteria	Weighting Factor	Description
Manufacturability	5	Feasibility, the ability to be integrated with the HPWH, and relative manufacturing cost.
Serviceability	5	Field replacement/repair of the CMS.
Reliability	5	The ability to evaporate condensate repeatedly and consistently and independent of the HPWH system.
Off-Cycle Evaporation	5	The ability to remove condensate when the HPWH is off.
Ability To Store Condensate	3	The ability to store condensate when the HPWH is on.
Re-evaporation Environment	3	The environment at which the condensate re-enters the air.

Envelope Impact	3	The effect on the overall size of the system.

3.4.11. Results of Conceptual Evaluations

Table 3-22 shows that both concepts are equally attractive. An additional container along the side of the tank entails more manufacturability and serviceability concerns, and, though the concerns can be dampened by altering the design of the CMS, integration of a second reservoir is more involved compared to the integration of a single condensate drain pan. The Single Pan also allows for more flexibility in container capacity. However, the Along the Side Concept is favorable in consideration of re-evaporation environment, off-cycle evaporation ability, and reliability. In the case of a single drain pan, the re-evaporated condensate can be blown out of the shroud by the fan. Total blow out, however, is not guaranteed, unlike in the case of direct re-evaporation to the outside of the HPWH's shroud. The Along the Side of the Tank conceptual design allows for better off-cycle evaporation because it has more sensitive control capabilities.

Table 3-22: Conceptual Design Comparison Results

Criteria	Weighting Factor	Single Pan	Along the Side
Manufacturability	5	0	-
Serviceability	5	0	-
Reliability	5	0	0
Off-Cycle Evaporation	5	0	+
Ability To Store Condensate	3	0	-
Re-evaporation Environment	3	0	+
Envelope Impact	3	0	0
	Total	0	-5

3.4.12. Conclusions and Recommendations

The CMS design exercise presented two realistic and simple conceptual designs. Manufacturer preference (based on manufacturing and serviceability advantages) led us to select the Single Drain Pan Concept. However, both designs present reliability issues. The Single Drain Pan Concept requires two float switches (moving parts). The Along the Side of the Tank Concept has no moving parts, but uses a pressure switch, which may be susceptible to malfunction if exposed to liquid for a prolonged period of time. In addition, the air trap periodically needs to be vented to atmosphere. These reliability issues must be tracked carefully as prototype fabrication and testing proceed.

3.5. Evaporator Fan Selection

The evaporator fan selection process was performed as outlined in Figure 3-11. First, we determined the air-flow rate that minimized total heat-pump power draw. We then checked this air-flow rate to confirm that it would result in acceptable noise levels and condensate formation rates.

Step 1: Select Desired Air Flow Rate at EF Conditions	Characterize Fan Performance	→	Define Design Parameters	→	Minimize Total Power Draw	→	Check Noise Level	→	Calculate Condensate Formation in Worst Cases
Step 2: Perform Trade Study to Select Fan Configuration	Single Fan; Runs when HPWH is on		Double Fan; One turns off when ambient temp too high		Variable Speed Fan; Optimized for no condensate				

Figure 3-11: Fan Selection Approach

3.5.1. Characterize Fan Performance

To characterize fan performance, we correlated fan power and noise level as a function of air-flow rate (for free discharge) for a series of muffin fans typical of those used in earlier HPWH prototypes (see Figure 3-12). Noise level ranges from 45-60 dB, while power ranges from 15-40 W. Based on the data provided by manufacturers, straight-line curve fits seemed most appropriate, although in theory power draw should vary with the cube of air-flow rate.
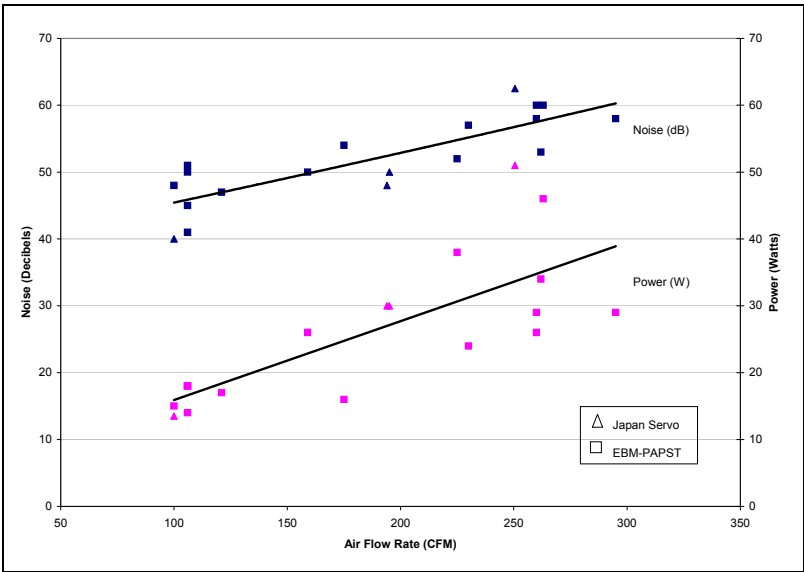
Figure 3-12: Typical Fan Noise and Power vs. Air Flow Rate

3.5.2. *Define Design Parameters*

Next we defined the design parameters that we needed to perform the fan air-flow rate optimization:

- Ambient air conditions;

- Evaporator model;

- Compressor efficiency; and

- Target noise level.

Ambient Air Conditions

We selected an ambient air condition of 70°F and 50% RH, corresponding to typical indoor conditions. Coincidentally, these conditions are similar to those required by the DOE Energy-Factor test procedure.

Evaporator Model

For the purposes of this analysis, we used a nominal evaporator capacity of 3000 Btuh, which is roughly consistent with the nominal condensing capacity design condition of 3600 Btuh. Using an evaporator performance computer model developed by Heatcraft (the evaporator manufacturer), we generated a plot of evaporating temperature and condensation formation rate as a function of air-flow rate through the evaporator (see Figure 3-13). The figure shows that no condensation forms at air-flow rates greater than 175 cfm.
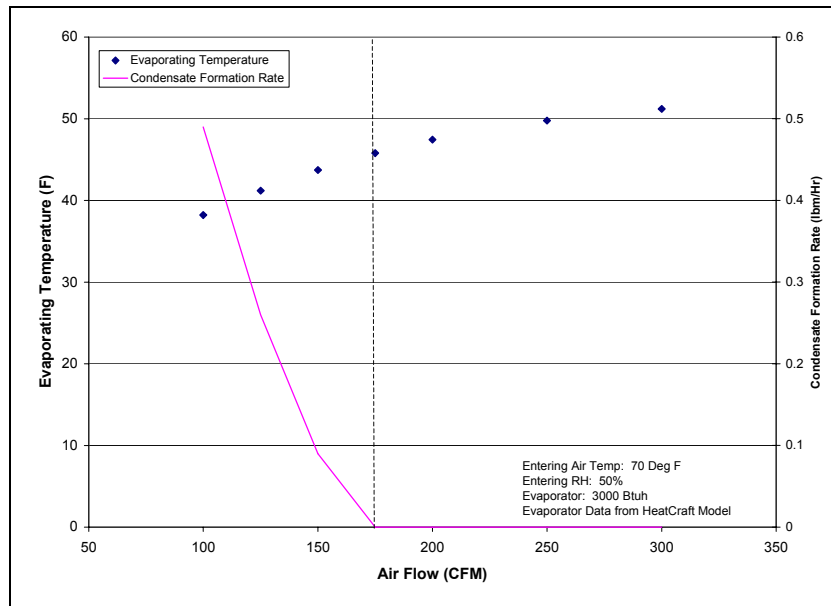
Figure 3-13: Evaporating Temperature vs. Air Flow Rate

Compressor Efficiency

We estimated compressor power (at a fixed condensing temperature of 110 °F and a 3000 Btuh evaporator) as a function of air-flow rate using a compressor manufacturer's performance map.

Noise Level

Figure 3-14 shows the noise levels generated by various household appliances. The noise levels generated by 100-to-300-cfm fans (shown previously in Figure 3-12**)** range from 45 to 60 dB. This corresponds to the noise levels generated by clothes dryers, down-draft cook tops, microwaves, and wall ovens. We judged that HPWH noise levels in this range would be acceptable to end users. (This contention, however, can only be confirmed through field testing and user acceptance evaluation.)
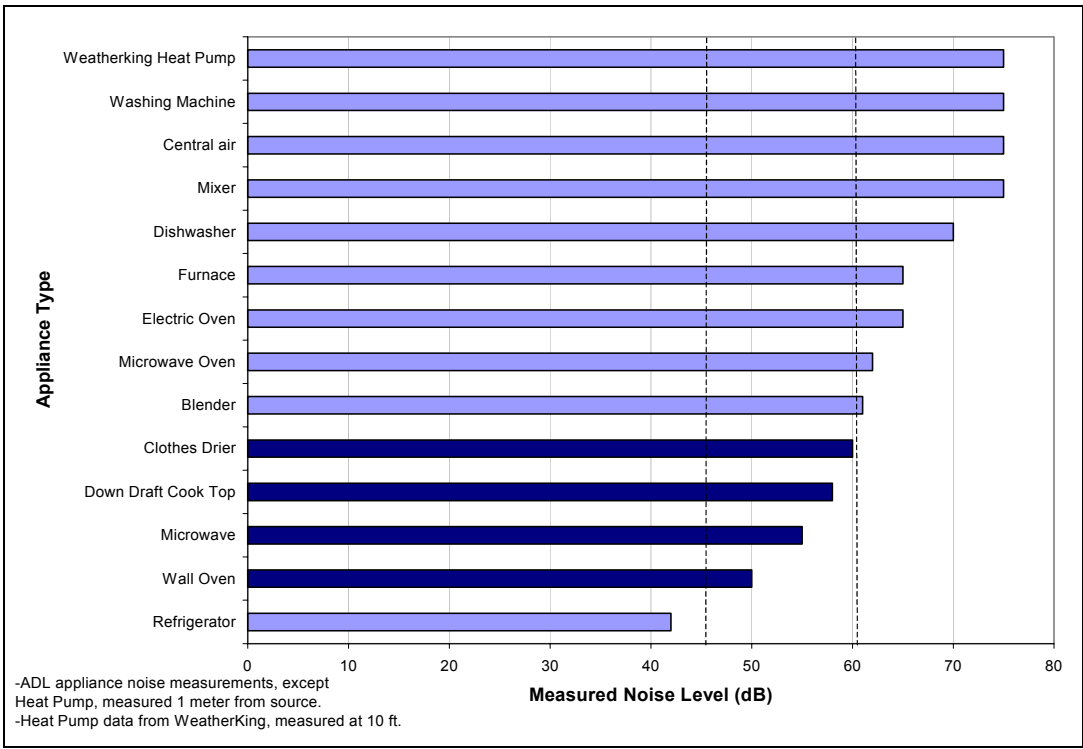
Figure 3-14: Appliance Noise Level Chart

3.1.8.3 Air-Flow Optimization

Figure 3-15 summarizes the power-draw estimates (for fan, condensate-evaporation, compressor, and total power draws) as a function of air-flow rate. The minimum total power draw occurs at an air-flow rate of 200 cfm. This air-flow rate corresponds to an estimated fan-power draw of 30 W. It results in no condensation formation for 70°F and 50% RH ambient air, and provides an acceptable noise level (estimated at about 52 dB).
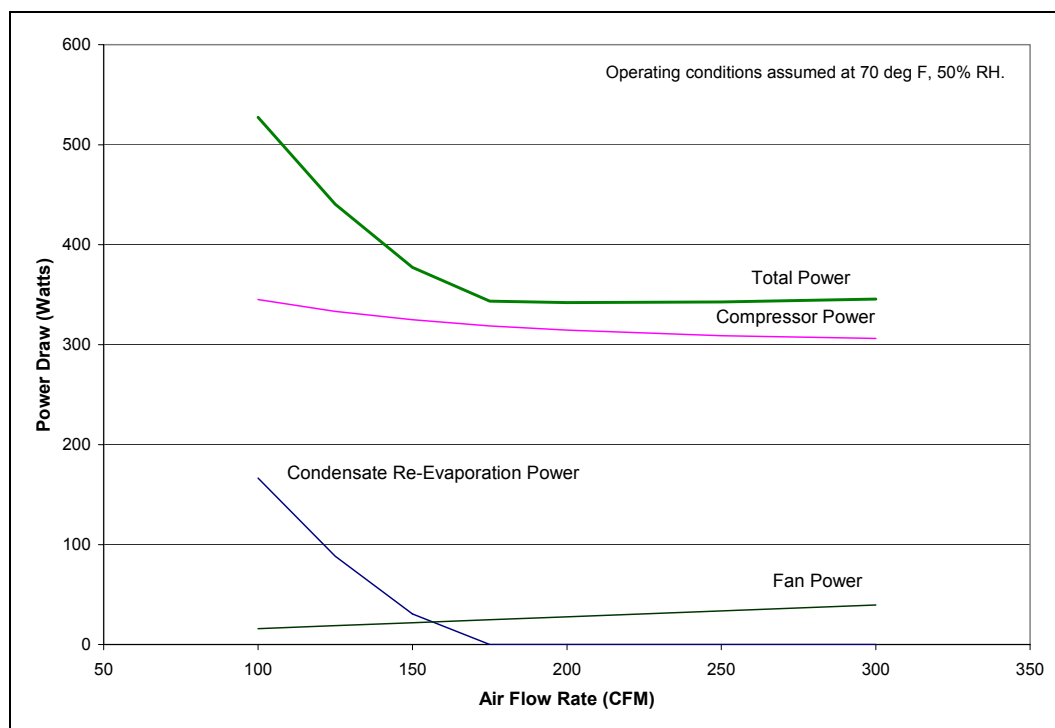
Figure 3-15: Power Draws vs. Air Flow Rate

Operation in Outdoor Installations

While 200 cfm of air flow results in no condensate formation in a 70 °F, 50% RH environment, these conditions do not reflect a worst-case condition. Some water heaters are installed in unconditioned shelters (such as garages), where high humidity conditions are more likely to occur. Table 3-23 lists the wet-humid design conditions for three California cities (along with a typical indoor condition). Eureka, CA represents a worst-case design condition in terms of tendency to form condensate.

Table 3-23: Design Conditions for Latent Loads

City	Dry Bulb Temp	Wet Bulb Temp
Eureka, CA	67°F	59°F
Los Angeles, CA	81°F	64°F
Victorville, CA	98°F	65°F
Closet Condition	55°F	52°F
Indoor Condition	70°F	58°F

Figure 3-16 shows total power draw at the wet-humid design condition for the three cities, and for the indoor condition. Higher dry-bulb temperatures produce higher evaporating temperatures and, hence, higher heating capacities and power draws. Significant in the figure is that the power draw for Eureka, CA does not level off before 300 cfm, suggesting that condensate forms (hence, requiring power for evaporation) even with air flows as high as 300 cfm.
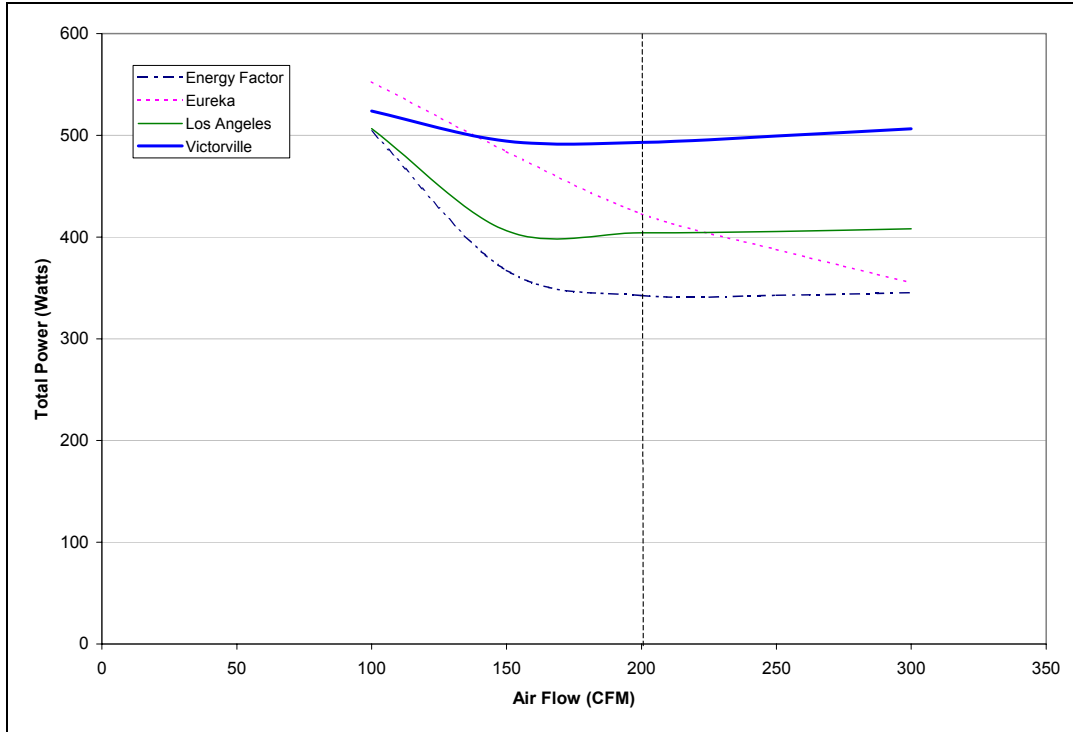
Figure 3-16: Fan Air-Flow Rate In Various Conditions

Table 3-24 lists condensation formation rates for outdoor installations in the three cities at the wet-humid design condition for 200 cfm air flow. Eureka, CA is the only city in which condensate forms at this condition. The maximum condensation formation rate (for installations in unconditioned spaces) is about 0.34 lbm/hour. This is safely below the condensation re-evaporation rate for which the CMS was designed.

Table 3-24: Comparison Cities and Conditions

City	Qe (Btuh)	Te (°F)	Condensate Generation (lb/hr)	COP	Total Power (watts)
Eureka, CA	2831	46.1	0.335	2.77	422.8
Los Angeles, CA	3742	53.6	0.000	2.91	404.1
Victorville, CA	4828	68.7	0.000	3.04	492.9
Closet Condition	2425	25.0	0.731	2.05	623.5

Indoor Condition	3000	47.3	0.000	2.79	322.5

3.5.3. Fan Configuration Evaluation

Having established the optimum air flow rate, we conducted a trade study to determine the best fan configuration. We evaluated three configurations:

- Single, fixed-speed fan;

- Two, fixed-speed fans; and

- Single, variable-speed fan.

Description of Alternatives

The single, fixed-speed fan offers only binary (on or off) operation. Two, fixed-speed fans provide the option for an intermediate air-flow rate (by operating a single fan). The variable-speed fan offers the greatest design flexibility, permitting operation over a range of air-flow rates.

3.5.4. Definition of Criteria and Weighting Factors

The three options were compared based on the criteria listed in Table 3-25.

Table 3-25: Fan Configuration Selection Criteria and Weight

Criteria	Weight	Definition
Cost	5	The relative manufactured cost.
System Reliability	5	The contribution to the overall system reliability.
Noise Level	5	The expected noise from the alternative's operation.
Power	3	The total power required to run the fans.
Air Distribution	3	The ability to provide a uniform distribution of air across the face of the evaporator coil.
Controls	1	The number and relative cost of the controls required for operation.
Manufacturability	1	The feasibility of the part to be manufactured and integrated into the HPWH.

Cost

Two fans cost more than one, even if the air-flow rate of each fan is half that of the single fan. However, variable-speed drives are very expensive, and a variable-speed fan will cost far more than two fixed-speed fans.

System Reliability

Two fans or a variable-speed fan (with the appropriate control system) additional offer protection in warm ambient. As the evaporating temperature approaches dangerous levels, the air-flow rate can be dropped, thereby lowering evaporator temperature without having to disable the heat-pump. A single, fixed-speed fan does not offer this protection at warm ambient without simply disabling the heat-pump.

Noise Level

Use of two fans will generally result in lower noise levels relative to a single fan as air velocities will tend to be lower. The noise level of a variable-speed fan would change with fan speed. However, some variable-speed drives make "whining" noises, which could be unacceptable. The variable-speed fan has no clear noise-level advantage relative to a single, fixed-speed fan.

Power

Two fans will tend to draw more power than one (as illustrated in Figure 3-12). The variable-speed fan should provide the lowest power draw, as air flow can be matched to operating conditions (with the proper control system). However, the variable-speed fan will draw slightly more power at full-speed operation relative to a single fan.

Air Distribution

Even air distribution is important to ensure optimum performance of the evaporator coil. Given the rectangular shape of the evaporator, more uniform air flow can be provided with two fans than with one.

Controls

The controls are a necessary for all three fan configurations. The controls for two fans, operated as described previously, are more complicated relative to a single fan. The variable-speed fan requires the most complicated controls to use it effectively.

Manufacturability

A dual-fan HPWH is slightly more difficult to manufacture than the single-fan HPWH. The variable-speed fan is probably equivalent to the single, fixed-speed fan in manufacturability.

3.5.5. Results and Conclusions

Table 3-26 shows results of the fan-configuration trade study. In this table, the single, fixed-speed fan served as the baseline, and the other two configurations were rated relative to the baseline. The two, fixed-speed fan configuration is the clear choice.

Table 3-26: Fan Configuration Decision Matrix

Criteria	Weight	One Fan	Two Fans	Variable Speed Fan
Cost	5	0	-1	-3
System Reliability	5	0	+2	0
Noise Level	5	0	+1	0
Power	3	0	-1	+1
Air Distribution	3	0	+1	0
Controls	1	0	-1	-2
Manufacturability	1	0	-1	0
	TOTAL	0	8	-14

3.6. HPWH Control System

The HPWH control system monitors system operation and performs specified actions based on the current condition of the water heater. Unlike an electric-resistance water heater, the addition of the heat-pump requires a more sophisticated control scheme. Three dominant factors, cost, reliability, and performance, affect control system selection and design. Two alternatives were considered:

• A conventional control scheme; and

• A micro-controller based system.

3.6.1. Control Specification

The control specification defines the operation of the control system. It is the basis of the design of the control circuit. There are two basic modes of operation: heat-pump and electric resistance. Table 3-27 describes HPWH modes of operation. Table 3-28 describes operation in the electric-resistance mode water heater.

Table 3-27: HPWH Modes of Operation

Condition	Operational Mode
Ambient Temp < 40°F	Operates in electric-resistance mode
40°F <= Ambient Temp < 90°F	Operates in heat-pump mode, both fans active
80°F <= Ambient Temp , = 90°F	Operates in heat-pump mode, with 1 fan active
90°F <= Ambient Temperature	Operates in electric-resistance mode
Condensate in Reservoir	Operate CMS
Condensate Reservoir Full	Operates in electric-resistance mode

Table 3-28: Electric-Resistance Water-Heater Temperature Ranges

Condition	Operational Mode
Upper Tstat< set point	Upper element on, Lower element off, CMS deactivated
Upper Tstat >= 135°F and Lower Tstat < set point	Upper element off, Lower element on, CMS deactivated
Upper Tstat and Lower Tstat >= set point	Both elements off
Condensate in Reservoir	Operate CMS

3.6.2. Conventional Control Scheme

The conventional control scheme utilizes relays and thermostats in the circuit to control the state of the system. The circuit configuration determines the control algorithm. Figure 3-17 shows the recommended control scheme based upon the control requirements. Table 3-29 provides a cost breakdown of the proposed control scheme.
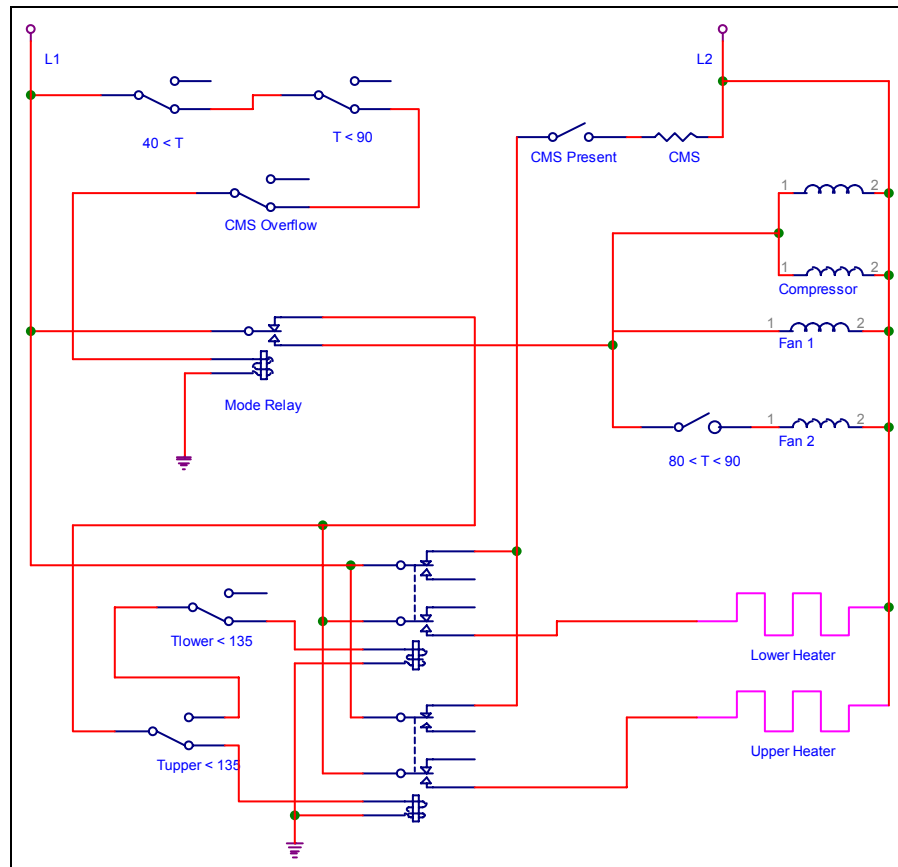

Figure 3-17: HPWH Control Diagram Using Conventional Controls

Table 3-29: Conventional Control Scheme Cost Analysis

Part	Unit Cost	Qty	Cost
Relay – Single Pole Double Throw	$ 5.50	1	$ 5.50
Relay – Double Pole Double Throw	$ 6.50	2	$ 13.00
CF/Overflow Switch	$ 1.00	2	$ 2.00
Thermostat	$ 2.00	4	$ 8.00
Thermostat	$ 2.75	1	$ 2.75
		TOTAL	$ 31.25

3.6.3. Micro-Controller Control Scheme

The micro-controller control scheme utilizes a micro-controller to perform the decision and relays for power routing. The control algorithm is based entirely in software so changes in the algorithm do not require hardware modification Figure 3-18. Shows the recommended control scheme based upon the functional requirements. Table 3-30 provides a cost breakdown of the proposed control scheme.
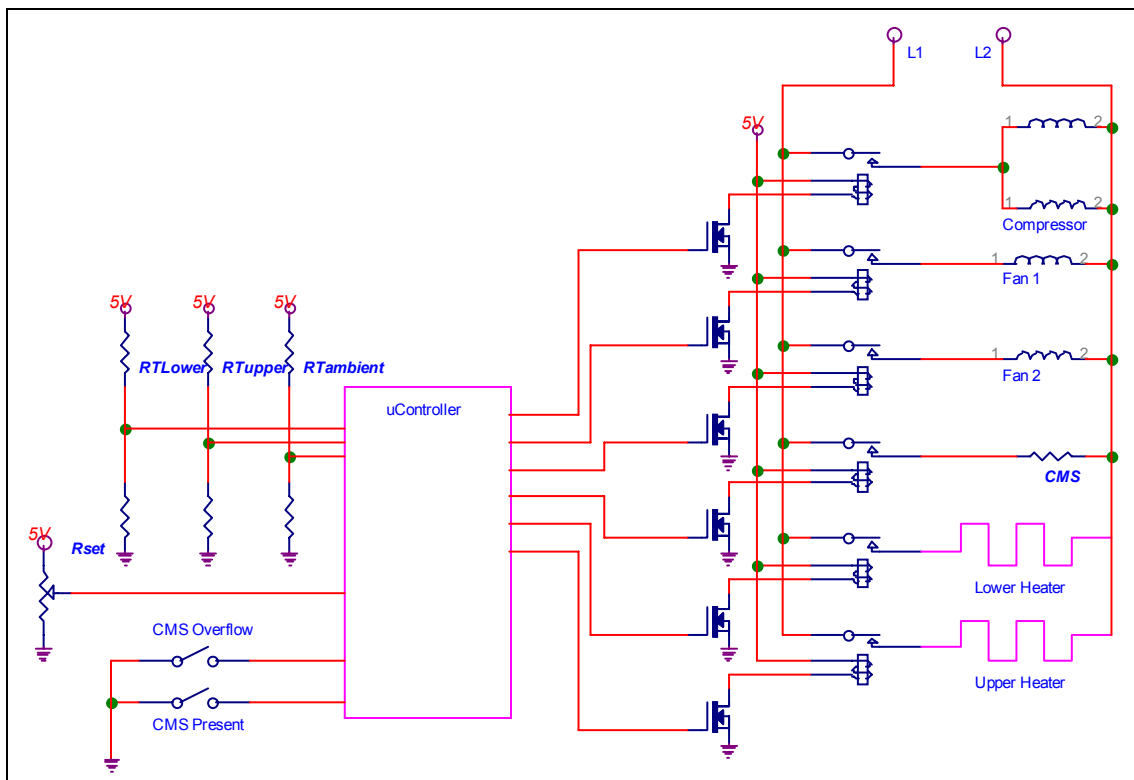

Figure 3-18: Micro-Controller Schematic Diagram

Table 3-30: Micro-Controller Control Scheme Cost Analysis

Part	Unit Cost	Qty	Cost
Micro-controller	$ 1.00	1	$ 1.00
PCB	$ 2.00	1	$ 2.00
FETs	$ 0.20	6	$ 1.20
Misc. Resistors	$0.50	1	$ 0.50
Regulator	$0.50	1	$0.50
Transformer	$ 2.00	1	$ 2.00
Relay – Single Pole Single Throw	$ 2.30	6	$ 13.80
CF/Overflow Switch	$ 1.00	2	$ 2.00
Thermistor	$ 0.50	3	$ 1.50
		TOTAL	$ 24.50

3.6.4. Recommendation

The HPWH requires a more intelligent control scheme than standard water heaters. The additional complexity is ideally suited to micro-controller based systems. The cost of equal systems is also in favor of the micro-controller control scheme, as power devices may be controlled with significantly lower voltages that are not available in the conventional control scheme. Simpler conventional schemes were considered, but many of the desirable features would not be possible and thus would reduce the performance and reliability of the system as a whole. The micro-controller control scheme will also allow for future modifications of the control algorithm with relative ease (if necessary), which may reduce time to market.

4.0   Component Testing

4.1. Contact Resistance

The contact resistance between the condenser tubing and the water tank is a large contributor to the HPWH's overall thermal resistance between the condensing temperature and the tank water temperatures. However, there is an insufficient amount of data to quantify the actual resistance.

The purpose of this experiment was to determine the thermal resistances between the copper tubing and the steel tank when using four different conductive media: no media, thermal adhesive tape, Thermal Mastic, and solder. The resistances represent the extremes for the alternative media.

- Section 4.1.1 of the report describes the experimental apparatus and procedure;

- Section 4.1.2 describes the thermal resistance model and assumptions required for calculating the contact resistance for each media;

- Section 4.1.3 contains the experimental results; and

- Finally, Section 4.1.4 provides a conclusion.

4.1.1. Experimental Setup

The experimental apparatus, as seen in Figure 4-1, is a scaled down model of the HPWH's tank and condensing coils. For each of the test cases, (described below), the copper tubing and heating element assembly was wrapped around a steel container. After attaching the tube, high temperature and low temperature insulation were wrapped around the entire system.

Prior to powering the resistive heater and recording data, the container was filled with water. Water was flowed into and out of the cylindrical container to maintain a relatively constant water temperature. Eight temperatures were recorded during each test. Figure 4-2 shows the placement, (common for each test), of the thermocouples.
Figure 4-3 illustrates the specific location of the thermocouple relative to the outer surface of the tubing and the inner surface of the container; T_o and T_i represent the exterior and interior temperatures, respectively.

Bare Tank

Figure 4-1: Experimental Apparatus

For the bare tank test case, d-shaped copper tubing was wrapped around the steel shell. Each wrap was spaced approximately 1" apart, as seen in Figure 4-2. The upper spiral of the tubing was then clamped to steel to ensure that no slippage occurred during data acquisition. This hand wrapping process was repeated in each of the other three test cases. The other cases, however, involved an extra step for the application of the contact conductive media.

Thermal Tape

Prior to attaching the d-shaped cooper coil to the container, the thermal tape was wrapped around the shell. The tape covered the contact area between the tubing and the shell. The top two-thirds of the section was wrapped in a 10 mil version of 3M Thermally-Conductive Adhesive Transfer Tape, while the bottom third was wrapped in a 5 mil version.

Thermal Mastic

During the set up for the Thermal Mastic test case, a layer of the conductive paste, Thermal Mastic Temperature Transfer Compound Q77-1200, was applied to the container. The layer spanned the height of the coiled tubing. After the paste had been applied, the d-shaped copper tubing was attached to the container.

Soldering

Unlike in the other three test cases, normal shaped ¼"copper tubing was wrapped around the container. The container's rust preventive coating was stripped off prior to the wrapping the tubing in order to ease the soldering process. No clamps were necessary, as the soldering provided enough bond for the copper tubing.
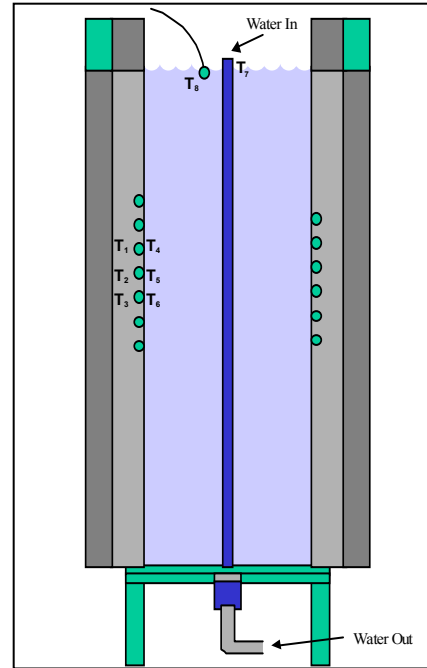

Figure 4-2: Test Setup

4.1.2. Theoretical Model

The heat transfer between the copper tubing, the conductive media, and the steel cylindrical container is solely dependent on the temperature differences between the surfaces. Using thermal resistance, we modeled the heat transfer from the outer surface of the copper tubing to the inner surface of the container. The overall heat transfer rate, Q, is defined by the equation:

$$Q = \frac{\Delta T}{\sum R} . \qquad (3.1)$$

Figure 4-3: Thermocouple Placement

Where $\sum R$ represents the overall thermal resistance and is equal to the inverse of the overall heat transfer coefficient and ΔT is the temperature difference between the copper tubing and the inner surface of the container. The thermal circuit consists of two resistances: resistance due to conduction through the thickness of the steel cylinder, R_{cond}, and the contact resistance of the conductive media, R_{cr}. The conductive resistance is defined by:

$$R_{cond} = \frac{L_s}{(k_s A_s)} . \qquad (3.2)$$

Where L_s represents the thickness of the steel container, A_s represents the contact area between the conductive media and the container, and k_s is the thermal conductivity of steel.
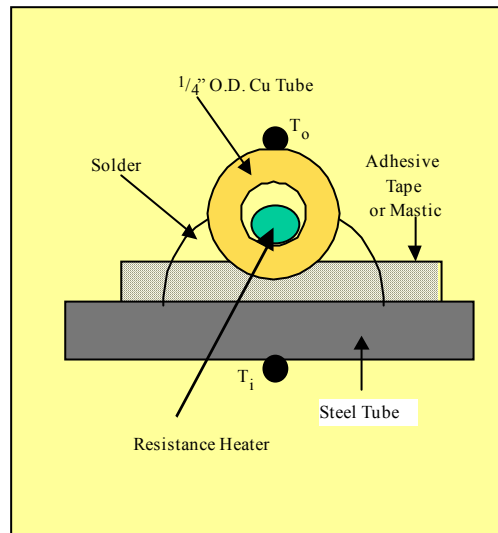
Substituting the definition of conductive resistance into the equation for the heat transfer rate and solving for R_{cr}:

$$R_{cr} = \frac{\Delta T}{Q} - \frac{L_s}{(k_s A_s)}.$$

(3.3)

4.1.3. Results

For each test case, the eight temperatures and the power draw of the heating element were recorded. Assuming the heat transfer rate for the system to be equal to the power input to the resistive heating element, and ΔT be the difference between the average internal temperatures and the average external temperatures, the contact resistance was calculated for each test. Table 4-1 summarizes the steady state results for the different media.

Table 4-1: Contact Resistance Experiment Summary of Results

Media	Bare Tube	Thermal Tape[1]	Thermal Mastic	Solder
Avg. T_o (^{o}F)	125	178	125	97
Avg. T_I (^{o}F)	74	83	91	87
Q (Btu/h)	427	949	1670	1630
Unit Length Contact Rsistance (^{o}F-ft/Btuh) [2]	0.396	0.352	0.066	0.022
Contact Resistance (^{o}F/Btuh)	0.036	0.032	0.006	0.002

[1]The thermal contact resistances for the thick and thin version were approximately 0.033 ^{o}F/Btuh and 0.030 ^{o}F/Btuh, respectively.

[2] The length of tubing was 11 ft.

4.1.4. Conclusion

The experimental results provide order of magnitude differences between the contact resistances for the four conductive media. While the Thermal Mastic has a resistance a factor of 3 larger than that of solder, the Thermal Mastic's resistance is still 83% lower than the resistance for the Bare Tube.

4.2. Experimental Verification of Expansion-Device Evaluation

The evaluation of expansion-device alternatives suggests that both the capillary tube and thermostatic expansion valve (TXV) are similarly attractive (with the TXV perhaps having a slight edge). To confirm the performance characteristics of the three expansion-device options, we conducted a laboratory test of the three expansion devices installed in otherwise identical prototype HPWHs. Both the capillary tube and the automatic expansion valve (AXV) utilized a suction-line accumulator, but the TXV did not.

The tests were conducted in EMI's environmental chamber, using Lab-View software for data acquisition. Tank water temperatures were measured using a thermocouple "tree" having six thermocouples along its length. The thermocouple "tree" was inserted into the top of the tank, and measured water temperatures near the centerline of the tank at six different elevations. The average of the six temperatures was used to determine tank water temperature rise (and the corresponding heat-pump capacity).

We attempted to optimize refrigerant charge for the three test configurations.

The conditions for each test were:

- 70°F, 50% RH ambient;

- No resistance water heating used (heat-pump heating only);

- Water tank is filled completely with 60°F to 65°F water prior to the start of the test;

- The heat-pump is operated with no water draws until the average tank temperature exceeds 120°F; and

- Data are recorded automatically for the duration of the test.

Figure 4-4 shows the resulting average tank-water-temperature profiles, along with the calculated average heating capacity. The results show that the TXV provided the highest average heating capacity (4535 Btuh), 20 percent higher relative to the AXV and almost 30 percent higher relative to the capillary tube. These capacity differences are consistent with expectations based on the operating characteristics of each expansion device.

Since refrigerant charge was not optimized, both the TXV unit and the AXV unit displayed excessive subcooling at the condenser outlet. While this has only a secondary effect on capacity, it does prevent meaningful comparisons of compressor discharge temperatures (interesting from a system-reliability standpoint) and heat-pump power draws. Furthermore, there was evidence that the 20"-long, 0.040"-diameter capillary tube was not optimum. A slightly longer (more restrictive) capillary tube may have produced a somewhat higher heating capacity.
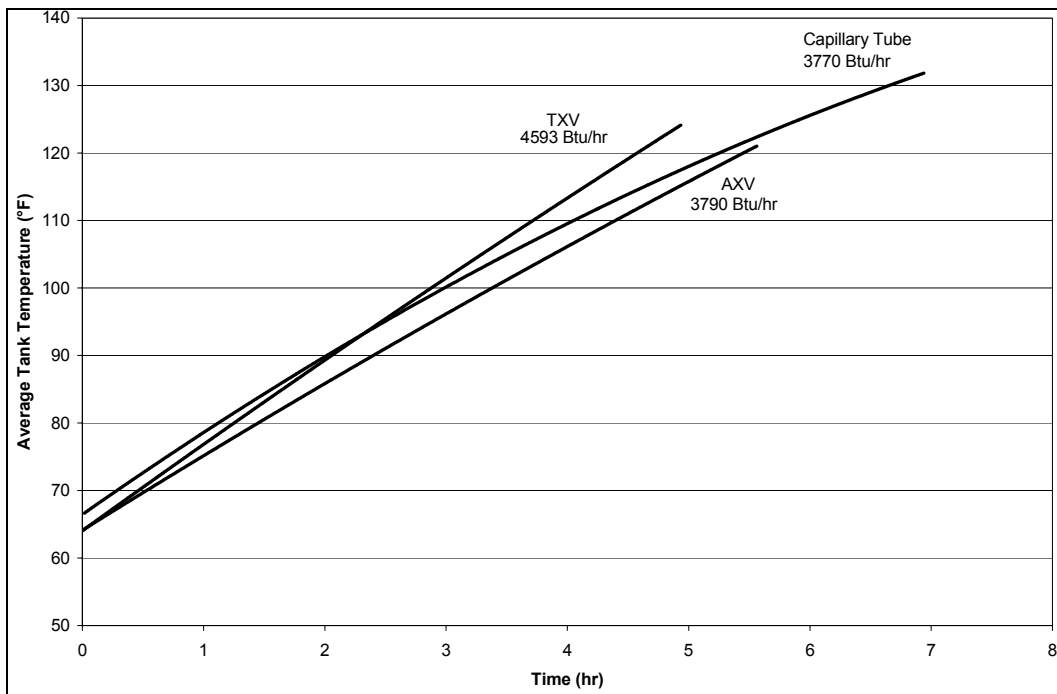
Figure 4-4: Comparison of Average Tank Temperatures vs. Time

Endnotes

[1] Andersson, Brandt, et. al.; *Aggregation of U.S. Population Centers Using Climate Parameters Related to Building Energy Use*; May 1986; Table 4.

[2] A 30 in. x 30 in. x 8 ft. high closet with 2x4 wooden frame and 5/8 in. gypsum board on the interior and exterior walls. No interior insulation. The exterior ambient temperature was assumed at 70°F.

[3] The evaporating temperature was assumed at 30°F below the interior closet temperature, with the condensing temperature held constant at 110°F.

[4] Using ¼ in. tube, with a 0.15 in. contact width with 1 in. spacing between wraps.

Appendix I – Compressor Capacity Solution

Given: Determine the maximum evaporating capacity as a function of closet temperature

Solution: Complete the following table.

Tcloset	40	45	50	55	60
Tevap	10	15	20	25	30
%FF10	215%	139%	94%	63%	41%
Pc	741	476	324	222	148
EER	4.3	5.3	6.1	6.9	7.6
Qe	3220	2521	1991	1538	1125
Qc	4914	3611	2732	2046	1464

Assumptions Used:
1) Condensing temperature is fixed at 110 F
2) Evaporating temperature is 30 F below closet temperature.
3) Heat loss from the tank is equal to 10% of the heating capacity.
4) Heat loss from the compressor is equal to 1/3 the input power.
5) Air exchange rate is equal to 1 air-change/hour (estimate).
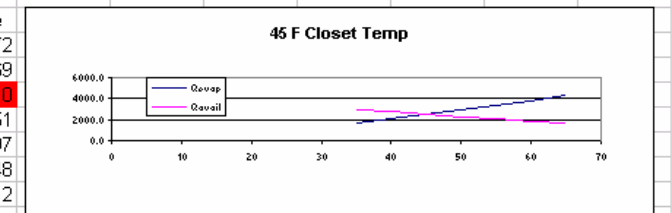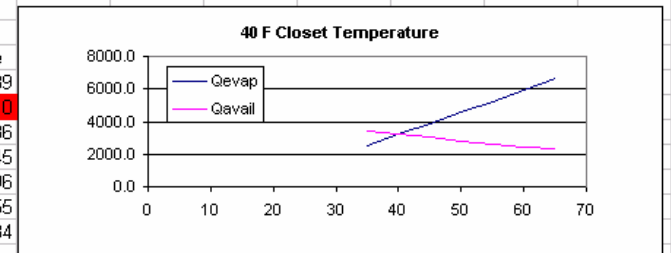6) Closet height is 8 ft.
7) Closet Wall Overall HTX = 0.55 Btu/hr*F*ft^2

Needed for Analysis:
1) EER (cooling) as a function of evaporating eemperature at constant condensing temperature
2) Power as a function of evaporating temperature at constant condensing temperature

Solution:

	EER as a function of Te is calculated as	-0.0018x2 + 0.234x + 2.1868
	Power as a function of Te is calculated as	0.0863x2 - 2.5481x + 361.41

Ambient Temperature:	75	F	Evap dT:	30	F
Wall Height:	8	ft	% Power Loss:	33%	%
Wall width:	3	ft			
Closet Wall Area:	96	ft^2	U:	0.55	Btu/hr*ft^2*F
Closet Volume:	72	ft^3	UA:	52.8	Btu/hr*F

MINIMIZATION GOAL
0

40F Closet temperature | Power | 215%

HPWH						Closet					Difference
T(closet)	T(evap)	EER	Pcomp	Qevap	Qcond	Qavail	Qwall	Qach	Qpower	Qtank	
35	5	3.31	754.2	2497.8	4222.5	3437	2112	53	849	422	939
40	10	4.35	740.7	3219.9	4913.7	3220	1848	46	834	491	0
45	15	5.29	736.5	3897.7	5581.9	3011	1584	40	830	558	-886
50	20	6.15	741.6	4558.6	6254.5	2814	1320	33	835	625	-1745
55	25	6.91	756.0	5225.2	6953.9	2629	1056	26	851	695	-2596
60	30	7.59	779.6	5914.7	7697.5	2460	792	20	878	770	-3455
65	35	8.17	812.5	6639.7	8497.7	2306	528	13	915	850	-4334

40 F Closet Temperature (Qevap, Qavail)

45F closet temperature | Power | 139%

HPWH						Closet					Difference
T(closet)	T(evap)	EER	Pcomp	Qevap	Qcond	Qavail	Qwall	Qach	Qpower	Qtank	
35	5	3.31	487.9	1615.8	2731.5	2987	2112	53	550	273	1372
40	10	4.35	479.2	2082.9	3178.6	2752	1848	46	540	318	669
45	15	5.29	476.5	2521.3	3610.8	2521	1584	40	537	361	0
50	20	6.15	479.7	2948.9	4045.9	2298	1320	33	540	405	-651
55	25	6.91	489.0	3380.1	4498.3	2083	1056	26	551	450	-1297
60	30	7.59	504.3	3826.1	4979.3	1878	792	20	568	498	-1948
65	35	8.17	525.6	4295.1	5497.0	1683	528	13	592	550	-2612

45 F Closet Temp (Qevap, Qavail)

50F closet temperature | Power | 94%

HPWH

T(closet)	T(evap)	EER	Pcomp	Qevap	Qcond		Closet Qavail	Qwall	Qach	Qpower	Qtank		Difference
35	5	3.31	329.4	1090.9	1844.2		2720	2112	53	371	184		1629
40	10	4.35	323.5	1406.3	2146.1		2473	1848	46	364	215		1067
45	15	5.29	321.7	1702.3	2437.9		2230	1584	40	362	244		527
50	20	6.15	323.9	1991.0	2731.7		1991	1320	33	365	273		0
55	25	6.91	330.2	2282.1	3037.1		1758	1056	26	372	304		-524
60	30	7.59	340.5	2583.3	3361.9		1531	792	20	383	336		-1052
65	35	8.17	354.9	2899.9	3711.4		1312	528	13	400	371		-1588
70	40	8.67	373.3	3235.2	4088.8								

55F closet temperature   Power   63%

HPWH

T(closet)	T(evap)	EER	Pcomp	Qevap	Qcond		Closet Qavail	Qwall	Qach	Qpower	Qtank		Difference
35	5	3.31	221.9	735.0	1242.5		2539	2112	53	250	124		1804
40	10	4.35	218.0	947.5	1445.9		2284	1848	46	246	145		1337
45	15	5.29	216.7	1146.9	1642.6		2032	1584	40	244	164		885
50	20	6.15	218.2	1341.4	1840.5		1783	1320	33	246	184		441
55	25	6.91	222.5	1537.6	2046.3		1538	1056	26	251	205		0
60	30	7.59	229.4	1740.5	2265.1		1297	792	20	258	227		-444
65	35	8.17	239.1	1953.8	2500.6		1061	528	13	269	250		-893

60F closet temperature   Power   41%

HPWH

T(closet)	T(evap)	EER	Pcomp	Qevap	Qcond		Closet Qavail	Qwall	Qach	Qpower	Qtank		Difference
35	5	3.31	143.5	475.2	803.3		2407	2112	53	162	80		1932
40	10	4.35	140.9	612.6	934.9		2146	1848	46	159	93		1534
45	15	5.29	140.1	741.5	1062.0		1888	1584	40	158	106		1146
50	20	6.15	141.1	867.3	1189.9		1631	1320	33	159	119		764
55	25	6.91	143.8	994.1	1323.0		1377	1056	26	162	132		383
60	30	7.59	148.3	1125.3	1464.5		1125	792	20	167	146		0
65	35	8.17	154.6	1263.2	1616.7		877	528	13	174	162		-386

Appendix II – Condenser Design

Given: Using the following design parameters, determine the heating capacity needed to prevent the upper element from energizing.

Parameters:

Starting Temp	135	Tank Diam	1.52 ft
with 17 db	118	Division Ht	0.42 ft
		Div. Volume	0.75 ft3
		Div. Mass	42.95

Solution: Using the condenser heat flux distribution, iterate until the estimated capacity it sufficient to provide 118 F water in the third bin from the top (30-35 in.)

Est. Capacity   **3952** <-- Iterative Solution

Condenser Heat Flux Distribution (Estimated from observation and experience)

	1st Draw	2nd Draw	3rd Draw	4th Draw	5th Draw	6th Draw
40-45						
35-40						
30-35						
25-30						
20-25						
15-20	0		0.304	0.352	0.352	0.352
10-15	0	0.304	0.304	0.304	0.304	0.304
5-10	0.652	0.348	0.300	0.304	0.304	0.304
0-5	0.348	0.348	0.092	0.04	0.04	0.04
	1	1	1	1	1	1

Condenser Capacity (calcuated from heat flux distribution and estimated capacity)

	1st Draw	2nd Draw	3rd Draw	4th Draw	5th Draw	6th Draw
40-45	0	0	0	0	0	0
35-40	0	0	0	0	0	0
30-35	0	0	0	0	0	0
25-30	0	0	0	0	0	0
20-25	0	0	0	0	0	0
15-20	0	0	1201.546	1391.264	1391.264	1391.264
10-15	0	1201.546	1201.546	1201.546	1201.546	1201.546
5-10	2577	1375.454	1185.7362	1201.546	1201.546	1201.546
0-5	1375.454	1375.454	363.62577	158.0982	158.0982	158.0982

Resulting Water Temperatures (on an hourly basis, from above heating capacities)

	1st Draw	2nd Draw	3rd Draw	4th Draw	5th Draw	6th Draw	
40-45	135	135.0	135.0	135.0	118.0	118.0	
35-40	135	135.0	135.0	118.0	118.0	118.0	
30-35	135	135.0	135.0	118.0	118.0	118.4	
25-30	135	135.0	118.0	118.0	118.0	94.4	
20-25	135	135.0	118.0	118.0	94.4	89.7	
15-20	135	118.0	90.0	85.6	86.0	86.0	
10-15	135	90.0	90.0	66.5	61.7	61.7	
5-10	58	58	58	58	58	58	
0-5	58	58	58	58	58	58	
average	58.0	68.7	74.0	67.0	65.9	65.9	66.6

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Consulting Services in Thermal/Fluid Technologies

September 8, 1999

Mr. Richard C. Williams
Consultant
Arthur D. Little, Inc.
20 Acorn Park
Cambridge MA 02140-2390

Re: Heat pump water heater condenser analysis results

Dear Richard:

Results of condenser design calculations are summarized below. The results indicate that the parameters that most affect condenser tube length are the interface configuration and overall temperature difference. Condenser lengths as short as 20 to 25 feet were estimated for the case of a soldered tube and a temperature difference of 25°F. Condenser tube lengths of over 400 feet were estimated for the case of a bare tube and a 5°F temperature difference.

The results indicate that a tube length reduction of 50 percent can be achieved relative to the estimated bare tube length through the use of an interface material such as epoxy. Once a reasonable interface condition has been achieved, additional improvements in the interface conductance result in additional tube length reductions of 20 to 25 percent.

Tube diameter and contact width have little effect on condenser tube length. The most important benefit of the larger tube diameter was a refrigerant-side pressure drop of only 25 percent of the smaller tube. Refrigerant-side pressure drops ranged from a few psi to as high as 125 psi.

Discussion

A simple two-dimensional thermal model was developed to estimate the overall thermal conductance between the condensing refrigerant and hot water. The thermal conductance was used to estimate the required condenser tube length for a specified heat rejection rate and overall temperature difference. The model considered the following elements:

w *Refrigerant-tube heat transfer*: This was modeled using estimated refrigerant heat transfer coefficients acting over the tube surface. The tube surface was divided into two regimes: the flat surface associated with the contact width and the remainder of the tube wall which acts as a fin. The heat transfer coefficient was estimated to be 600 Btu/hr-ft^2-°F. The thermal conductivity of the copper tube wall was set at 220 Btu/hr-ft-°F. In all calculations the tube wall thickness was set at 0.035 inches.

w *Contact resistance*: The contact resistance was modeled assuming an interface material of a specified thickness and thermal conductivity. The bare tube contact resistance was estimated using data for metallic surfaces in contact, which ranges from about 250 to 500 Btu/hr-ft^2-°F. Interface resistances are summarized in the following table.

Interface Condition	Assumed thickness mils	Thermal conductivity Btu/hr-ft-°F	Resulting Interface Conductance Btu/hr-ft^2-°F
bare	NA	NA	250
epoxy	1	0.12	1,440
mastic	1	0.5	6,000
solder	1	10	120,000

w *Tank wall*: The steel tank wall was divided into three segments: one in which conduction heat transfer flows perpendicular to the wall and upper and lower segments that behave like fins. The thermal conductivity was set at 25 Btu/hr-ft-°F and the wall thickness was set at 0.125 inches.

w *Glass liner*: The glass liner was treated as a stagnant layer of fluid with a heat transfer coefficient equivalent to the thermal conductivity divided by the liner thickness. For these calculations the thermal conductivity was specified to be 0.65 Btu/hr-ft-°F and the thickness was specified to be 3 mils. Note that the glass liner actually behaves as a fin in parallel with the tank wall. However, the axial conductance is negligible compared to the axial conductance associated with the steel tank wall, resulting in a negligible contribution to the heat transfer along the wall.

w *Water-side heat transfer*: This was modeled assuming natural convection heat transfer on a vertical heated wall. The temperature difference was calculated using the average temperature of the tank wall and the 90°F water.

w *Refrigerant pressure drop*: The refrigerant pressure drop was estimated using correlations for two-phase pressure drop in the condensing regime. The refrigerant mass flow rate was set at 46 lb$_m$/hr, based on a 3,600 Btu/hr capacity for the FF10HBK compressor.

Model calculations were performed in spreadsheet form. An iterative procedure was required to achieve convergence between the assumed tank wall temperature and the calculated water-side heat transfer coefficient.

Results

The modeling results are summarized in Tables 1 through 3 attached. Required condenser tube lengths were estimated for two tube diameters, several interface conditions, and three specified temperature differences for a required heat rejection rate of 3,600 Btu/hr. The average water temperature was set at 90°F. The results are summarized as follows:

w The two most important parameters are the interface condition and the allowable temperature difference between the condensing refrigerant and water. Tube lengths of order

w Tube lengths as short as 20 to 25 feet were estimated for the case of a soldered interface and a 25°F temperature difference.

w The allowable temperature difference has large effect on the required tube length, reducing the required tube length by more than a factor of five for a given configuration. Note that the tube length is not inversely proportional to temperature difference because of the effects that larger temperature differences have in increasing water-side heat transfer coefficients due to buoyancy effects.

w Once a reasonable interface conductance is established, the interface conductance has a small effect on the required tube length, as a soldered joint required only about 20 to 25 percent less tubing than an epoxy joint for a specified temperature difference.

w Tube diameter and contact width have the largest effect on tube length for the case of the poorest interface condition, i.e., the bare tube. Once a reasonable interface condition is established, the tube diameter and contact width affect the required tube length less than ten percent.

w Estimated refrigerant-side pressure drops ranged from as low as several psi to as high as 125 psi.

w The most important effect of tube diameter is one refrigerant pressure drop. The pressure drop for the 5/16 inch diameter tube was only about 25 percent that of the 0.25 inch diameter tube.

w For the values of tube spacing examined in this study (1 and 2 inches), the tube spacing has a small effect (10 to 15 percent) on the required tube length due to the phenomenon of decreasing fin efficiency associated with the tank wall.

These results represent the first step in the condenser design process. System-level design activities involving modeling and prototype testing will be necessary to determine the interaction between important condenser design variables (length, placement, etc.) and the refrigeration system and their impact on overall system performance.

Call me anytime if you have any questions or comments.

Sincerely,

Anthony F. Varone, Sc.D., P.E.
President
Thermodyne Inc.

Table 1: Required Condenser Tube Length, 0.25" Diameter Tube

Bond	Contact Conductanc	Required Length	Tube	Estimated Pressure

	e Btu/hr-ft^2-°F	@3,600 Btu/hr Capacity ft	Volume in^3	Drop psi
For all runs below: tube diameter = 0.25 inches, tube spacing = 1", refrigerant ht coefficient = 600 Btu/hr-ft^2-°F, average water temperature = 90°F				
0.15" contact width				
delta T=5°F				
bare	250	438	134	125
epoxy	1,440	226	69	63
mastic	6,000	190	58	53
solder	120,000	179	55	49
delta T=15°F				
bare	250	134	41	34
epoxy	1,440	64	19	16
mastic	6,000	52	16	13
solder	120,000	48	15	12
delta T=25°F				
bare	250	77	24	17
epoxy	1,440	36	11	9.5
mastic	6,000	29	8.8	6.3
solder	120,000	27	8.1	5.8
0.1875" contact width				
delta T=5°F				
bare	250	386	118	110
epoxy	1,440	213	65	60
mastic	6,000	185	56	51
solder	120,000	176	54	49
delta T=15°F				
bare	250	116	35	29
epoxy	1,440	60	18	15
mastic	6,000	50	15	12
solder	120,000	47	14	11.7
delta T=25°F				
bare	250	67	20	15
epoxy	1,440	33	10	7.3
mastic	6,000	28	8.4	6
solder	120,000	26	7.9	5.7

Table 2: Required Condenser Tube Length, 5/16" Diameter Tube

Bond	Contact Conductance $Btu/hr\text{-}ft^2\text{-}°F$	Required Length @3,600 Btu/hr Capacity ft	Tube Volume in^3	Estimated Pressure Drop psi
For all runs below: tube diameter = 5/16 inches, tube spacing = 1" and the refrigerant ht coefficient = 600 $Btu/hr\text{-}ft^2\text{-}°F$, average water temperature = 90°F				
0.1875" contact width				
delta T=5°F				
bare	250	379	210	30
epoxy	1,440	207	115	16
mastic	6,000	177	98	14
solder	120,000	169	94	13
delta T=15°F				
bare	250	114	63	7.9
epoxy	1,440	57	32	4
mastic	6,000	48	27	3.4
solder	120,000	45	25	3.2
delta T=25°F				
bare	250	66	36	4
epoxy	1,440	32	18	2
mastic	6,000	26	15	1.7
solder	120,000	25	13.7	1.6
0.25" contact width				
delta T=5°F				
bare	250	325	180	25.6
epoxy	1,440	193	107	15.3
mastic	6,000	171	95	13.6
solder	120,000	165	91	13.1
delta T=15°F				
bare	250	96	53	6.7
epoxy	1,440	53	29	3.7
mastic	6,000	46	25	3.2
solder	120,000	44	24	3.1
delta T=25°F				
bare	250	55	30	3.4
epoxy	1,440	29	16	1.8
mastic	6,000	25	14	1.6
solder	120,000	24	13	1.5

Table 3: Effect of Tube Spacing on Required Condenser Tube Length

Bond	Contact Conductance Btu/hr-ft^2-°F	Required Length @3,600 Btu/hr Capacity ft		Required Length @3,600 Btu/hr Capacity ft	
For all runs below: the refrigerant ht coefficient = 600 Btu/hr-ft^2-°F, average water temperature = 90°F					
		tube diameter = 0.25" 0.15" contact width		tube diameter = 5/16" 0.1875" contact width	
		1" spacing	2" spacing	1" spacing	2" spacing
delta T=5°F					
bare	250	438	409	379	350
epoxy	1,440	226	200	207	182
mastic	6,000	190	165	177	153
solder	120,000	179	154	169	145
delta T=15°F					
bare	250	134	127	114	107
epoxy	1,440	64	58	57	52
mastic	6,000	52	47	48	43
solder	120,000	48	43	45	40
delta T=25°F					
bare	250	77	74	66	62
epoxy	1,440	36	33	32	29
mastic	6,000	29	26	26	24
solder	120,000	27	24	25	22
		tube diameter = 0.25" 0.1875" contact width		tube diameter = 5/16" 0.25" contact width	
		1" spacing	2" spacing	1" spacing	2" spacing
delta T=5°F					
bare	250	386	357	325	297
epoxy	1,440	213	188	193	168
mastic	6,000	185	160	171	147
solder	120,000	176	151	165	141
delta T=15°F					
bare	250	116	109	96	90
epoxy	1,440	60	54	53	48
mastic	6,000	50	45	46	41
solder	120,000	47	42	44	38
delta T=25°F					
bare	250	67	63	55	52
epoxy	1,440	33	30	29	27
mastic	6,000	28	25	25	23

solder	120,000	26	23	24	21

Attachment 1

Third-Generation Prototype Fabrication and Laboratory Test Report

Attachment to Final Report: Design Refinement and Demonstration of Market-Optimized Heat-Pump Water Heater

March 5, 2004

California Energy Commission

Public Interest Energy Research

Contract Number 500-98-028

Note: This attachment is **NOT** a stand-alone report. It should be interpreted only in the context of the final report from which it is referenced.

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1.0 Prototype HPWH Design

We developed the design for the third-generation prototype of the HPWH as described in Section 3.1 of the Final Report. At that time, two sub-systems of the HPWH were recognized as needing additional design refinement: the condensate management system and the control system. Subsequent to the meeting, initial laboratory testing of the HPWH revealed the need for further modifications to the refrigeration system.

1.1. Condensate Management System (Factory Option)

Figure 1-1 shows the revised design of the condensate management system (CMS). The CMS design uses a secondary pan, connected to the evaporator drain pan by the drain line, which collects the condensate. (The original design concept called for integrating the CMS with the evaporator drain pan.) An electric-resistance immersion heater integrated into the pan, evaporates the condensate. One float switch triggers the CMS heater after allowing a small amount of condensate to accumulate. This switch also deactivates the CMS heater when the condensate is evaporated. A second float switch protects the drain pan from overflowing if the CMS is, not functioning properly, and cannot re-evaporate condensate fast enough.
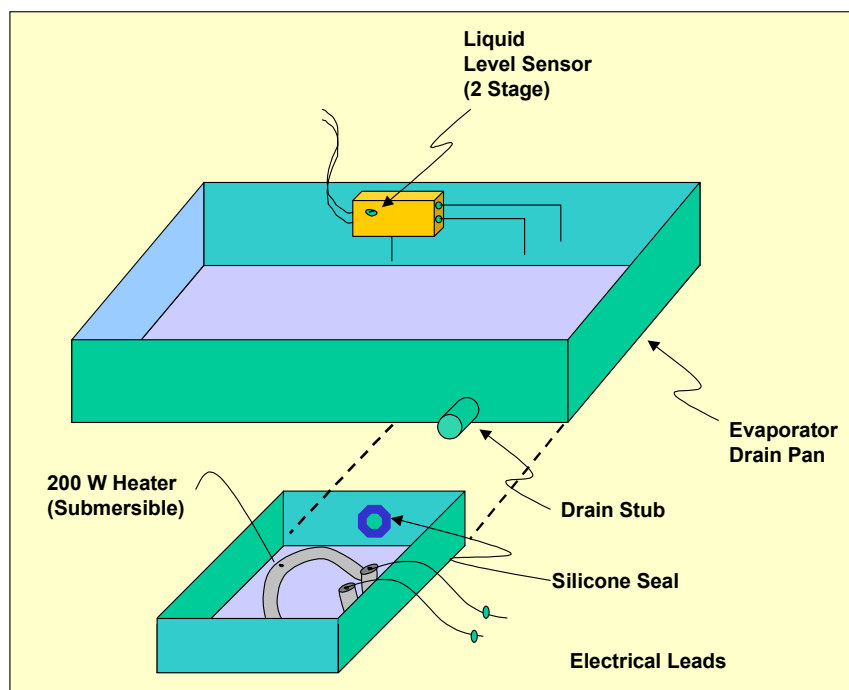
Figure 1-1: Prototype Model of Condensate Managment System

We learned from HPWH development and testing efforts prior to the current contract that it is advantageous to "store" a small amount of condensate under certain operating conditions. Specifically, in simulated-utility-closet tests, we observed a marked reduction in CMS energy consumption when a small amount of condensate was stored. Presumably, immediate re-evaporation of the condensate into the confined space of the closet resulted in repeated condensation and re-evaporation of condensate, unless some condensate was left unevaporated until after the heat pump shuts off.

With the concurrence of the Commission Contract Manager, we did not include the CMS in the prototype fabricated under this project.

1.2. Micro-Controller Development

The HPWH control system is based on micro-controller architecture. The micro-controller architecture utilizes a micro-controller to make decisions and operate relays for power routing. The control algorithm is based entirely in software, so changes in the algorithm do not require hardware modification.

This section describes the field-test version of the micro-controller. Although the production version will be functionally equivalent to the field test version, there are hardware design refinements needed to reduce the cost and increase the manufacturability. Those refinements are discussed below.

1.2.1.   Hardware Requirements

The field-test control system will have a terminal block that accepts the following analog and digital inputs cited in Table 1-1.

Table 1-1: Inputs to Micro-Control Board

Input	Device	Units
Dial Thermostat (User Input)	Var. Resistor[1]	Ω
Upper Water Temperature	Thermistor	Ω
Lower Water Temperature	Thermistor	Ω
Evaporator Temperature	Thermistor	Ω
Compressor Discharge Temp.	Thermistor	Ω
CMS Present	Float Switch	Open/Closed
CMS Overflow	Float Switch	Open/Closed

A HPWH Mode switch is also included on the printed circuit board. This will enable us to switch between HPWH mode and a conventional electric water heater. This switch is not accessible to the occupant.

The control system energizes power relays to control the following components as specified in the control sequence:

- Compressor;
- Fan 1;
- Fan 2;
- CMS Heater;
- Lower Element Resistance Heater; and
- Upper Element Resistance Heater.

Three LEDs are visible to the user and will indicate when the compressor, lower element or upper element operation. RS-232 and RJ-45 communication ports are also available for programming, de-bugging, and setting operating parameters.

The control system is powered from the same nominal 240 V, 60 Hz, single phase source as the HPWH. The power should be routed through a thermal switch to the control board and then to the controlled components. The power relays are rated for 240 V and 30 amps. A power terminal block is provided for connecting wires to the listed components.

The control hardware will incorporate measures that will prevent the upper element and lower element from energizing simultaneously. In addition, neither the compressor nor the CMS will energize when either element is energized.

The physical size of the prototype printed circuit board is approximately 5-in. wide by 7-in long and 1-1/2 in. high. There are four holes located on the corners of the board for stand-offs. There is no enclosure included.

1.2.2. HWPH Control Sequence

The software will control the HPWH. The software uses the control sequence, along with the values of the operating parameters, to determine which of the following two states the HPWH needs to operate in:

1. Heat-Pump-Water-Heater Mode; or
2. Electric-Resistance Mode.

The software variables are accessible via the serial communications port on the control board (RS-232). Table 1-2 summarizes the software variables, they are described in further detail within the context of the control sequence.

Table 1-2: Micro-Controller Software Variables

Variable	Description
HPWHSet	Set-point temperature of the HPWH.
TempLow	The lower evaporating temperature limit for heat-pump operation.
TempFan1	The evaporating temperature that shuts off one fan.
TempFan2	The evaporating temperature that shuts off both fans.
MinDisch	Minimum compressor discharge temperature that is required for continued operation of the HPWH after start-up.
MaxDisch	Maximum allowable compressor discharge temperature.
HysUpper	The deadband for the upper tank temperature
HysLower	The hysteresis for the lower tank temperature
HysMode	The hysteresis for other parameters, includes TempLow, TempFan1, and TempFan2
TempDiff	The lower tank temperature set-point temperature is lower than the HPWH set-point temperature by the amount of TempDiff.

Heat-Pump-Water-Heater Mode

The heat-pump-water-heater mode maximizes energy efficiency and is the preferred mode of operation. The control system is designed to maximize the use of the heat pump. Any time the lower tank temperature falls below the lower tank set-point temperature [HPWHSet – TempDiff – HysLower], the control system turns on the heat pump (compressor and fans).

Seven conditions will cause the control system to shut-off the heat pump (compressor and fans):

1. The lower tank temperature has reached the lower tank set-point temperature [HPWHSet – TempDiff].

2. The evaporating temperature is below [TempLow], at which point frost may begin developing on evaporator coils. The HPWH will re-start once the evaporating temperature reaches [TempLow + HysMode].

3. The upper tank temperature is less than the HPWH upper set-point temperature [HWPHSet – HysUpper], therefore the upper element is needed to heat the upper portion of the tank.

4. The lower tank temperature has reached the safe operating limit for the heat pump but the HPWH has not reached the HPWH set-point. In order to limit the discharge pressure, the HPWH is shut off. The lower element is energized until the HPWH set-point temperature is satisfied.

5. The CMS is at an overflow condition.

6. The HPWH Mode switch is closed.

7. Anytime the compressor discharge temperature is not between [MinDisch] and [MaxDisch] (within a short period after start-up), the HPWH will be switched to operate in the conventional electric water heater mode (assumes compressor has failed).

If the evaporating temperature becomes too high, the compressor is at risk of becoming overloaded (because the density of the suction gas increases). In order to prevent high evaporating temperatures the control system will follow the prescribed sequence:

1. When the evaporating temperature exceeds [TempFan1], one of the two evaporator fans is turned off. This causes the evaporating temperature to drop. The second fan will be turned on again once the evaporating temperature drops below [TempFan1 – HysMode].

2. If, after turning off one fan, the evaporating temperature exceeds [TempFan2], the second fan is also turned off. Natural convection then becomes the only mode of heat transfer (on the airside) for the evaporator. The single fan will be turned on again once the evaporating temperature drops below [TempFan2 – HysMode].

The above control sequence will increase the amount condensate generated by the evaporator (due to the evaporator becoming more latent).

An algorithm in the control system delays compressor restart for a short period after shutdown. This may create the situation where the lower element energizes (and remains on for the remaining delay period) if a water draw occurs immediately following the shutdown.

Electric-Resistance Mode

The upper element is energized any time the upper tank temperature is less than the HPWH set-point temperature [HPWHSet – HysUpper]. Regardless of the ambient conditions or the presence of condensate, the heat pump (or lower element) and CMS are shut-off and the upper element is energized. The upper element remains on until the upper tank temperature equals the HPWH set-point temperature [HPWHSet].

Five conditions will cause the control system to energize the lower element if the lower set-point temperature is not satisfied.

1. The evaporating temperature is below [TempLow], at which point frost may begin developing on evaporator coils.

2. The lower tank temperature has reached the safe limit for the heat pump, but the HPWH has not reached the HPWH set-point. In order to limit the discharge pressure, the HPWH is shut off. The lower element is energized until the HPWH set-point temperature is satisfied.

3. The CMS (if present) is at an overflow condition.

4. The HPWH Mode switch is closed.

5. Anytime the compressor discharge temperature is not between [Min Disch] and [Max Disch] (following a short period after start-up), the HPWH will be switched to operate in the conventional electric water heater mode (assumes compressor has failed).

Condensate Management System (Factory Option)

There are two levels that the CMS (if present) responds to — a condensate-present level and a condensate-overflow level.

When the CMS indicates there is condensate present (float switch is closed), the control system will energize the CMS heater (unless either of the electric elements are on). The heater will remain energized until the condensate float switch opens, indicating no more appreciable condensate. The CMS will operate during the HPWH cycle. However, if the HPWH is off and CMS is still required to evaporate condensate, one (1) fan will be turned on while the heater is on.

If the condensate reaches the overflow level, the HPWH will switch modes to a conventional electric water heater. Once the water heater has heated the water tank, the control system will energize the CMS heater and one (1) fan. The heater and fan will remain energized until the condensate present float switch opens, indicating that there is no longer any appreciable condensate.

The control system shall prevent the CMS heater and either of the electric resistance elements from being operated at the same time (to limit the current draw of the HPWH).

1.2.3. Power Wiring for the HPWH

The power wiring for the HPWH is shown in Figure 1-2. All power to the HPWH is initially routed through the electro-mechanical thermal switch (the same as found on a conventional electric water heater). The control board is powered from same source as the remaining components of the HPWH, as was set-forth in the design requirements. With the micro-controller architecture, each powered component has an associated relay which acts to switch the component on and off (depending upon the logic provided by the software). Power is directed through the relay to the component. Power interlocks for the components (which disallow the elements, compressor, or CMS to operate simultaneously) are provided in the hardware of the control board.
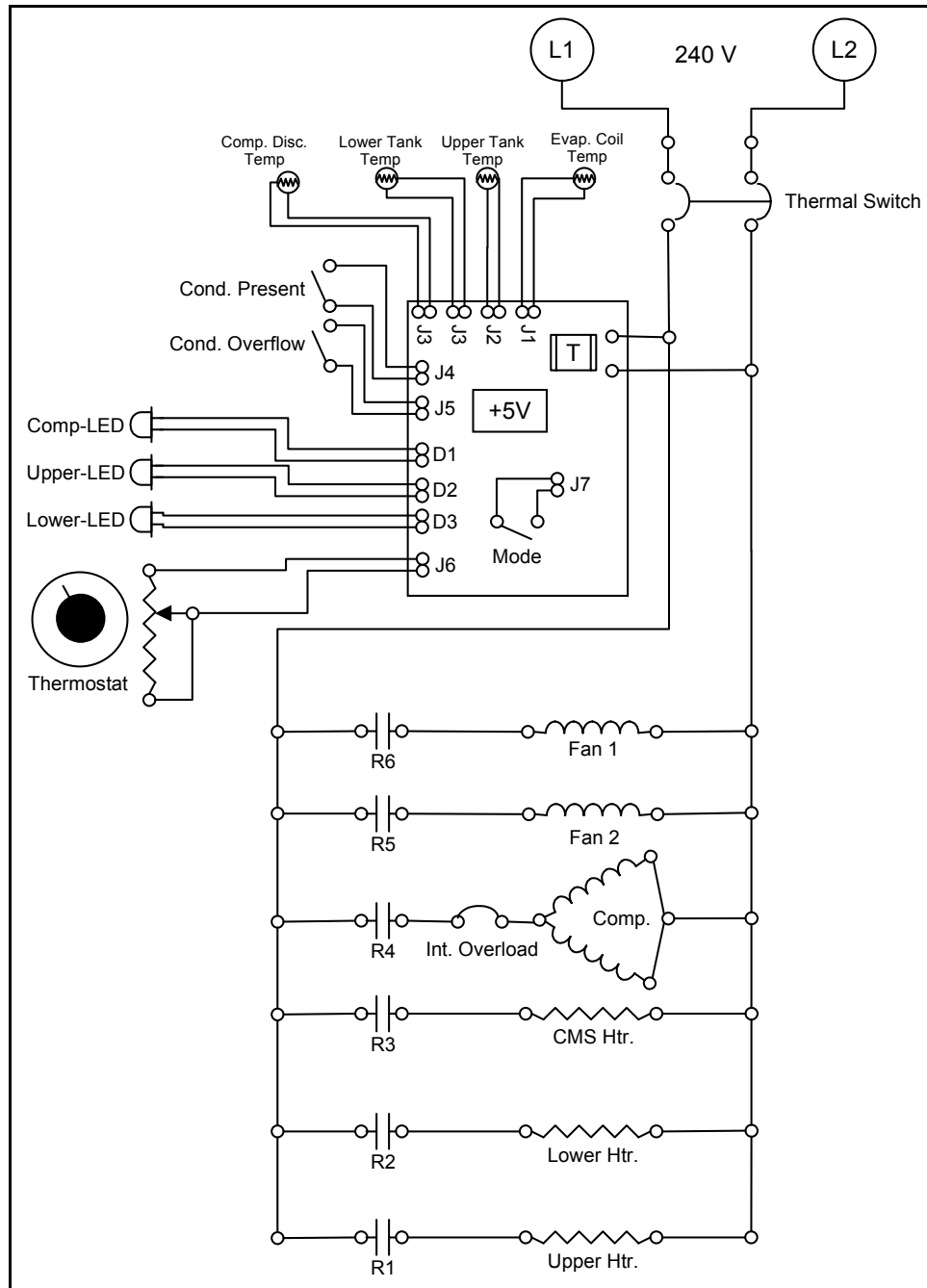
Figure 1-2: Power Wiring for HPWH

1.2.4. Production Version of the HPWH Control Board

There are significant hardware differences between the controller design recommended for the upcoming durability and field tests and the recommended production design. Table 1-3 highlights the major differences.

Table 1-3: Differences between the Field-Test and Production Versions of the HPWH Control Board

Field-Test Design	Production Design
Terminal blocks	Wiring harness and power routing on circuit board.
Hand placed components, wiring.	Designed for tape and reel* manufacturing and line assembly
Power relays	Power relays and Triacs
Programmable micro-controller chip	Fixed micro-controller chip
RS-232 and RJ-45 communication ports	No communication ports
240 V / 12 V AC Transformer	12 V off-line power source (still uses HPWH 240 V source)
Large, unprotected, printed circuit board	Smaller, protected, printed circuit board
HPWH Mode Switch	No switch
Conventional electric water thermal cutout switch	Thermal cutout switch selected for HPWH

The production design differs from the field-test version mainly as a way to reduce the costs of the control boards. Different components may be selected to reduce the cost and increase the manufacturability.

The reliability and occupant interface of these two designs are similar (other than the possible addition of LEDS). The field-test design also provides the flexibility to change the software and operating parameters. However, after the durability tests and field-tests are completed, the software and operating values should be fixed.

The reliability and occupant interface of these two designs are similar. The field-test design also provides the flexibility to change the software and operating parameters. However, after the durability tests and field-tests are completed, the software and operating values should be fixed.

1.3. Refrigeration System Enhancements

The U.S. Department of Energy, Oak Ridge National Laboratory (ORNL), conducted an initial Energy Factor test on the third-generation prototype. The resulting Energy Factor was 2.1, which was above our goal of 2.0 but below the level we expected. Further diagnostic testing and detailed analysis showed:

- A 5 psi to 7 psi pressure drop existed between the TXV and the suction of the compressor. The evaporator manufacturer estimates the evaporator pressure drop to be less than 1 psi. This condition caused an artificially low evaporating temperature, thus reducing the potential heating capacity of the HPWH.

- Significant thermal losses from the compressor shell as evidenced by:
  - The capacity of the HPWH was 60% to 75% of the estimated value. On average the heating capacity of the HPWH was approximately 3600 Btu/hr, but varied between 2000 Btu/hr and 4200 Btu/hr (depending on the water tank temperatures adjacent to the condenser). The manufacturer's compressor map suggests that we should have rejected between 4000 Btu/hr and 5000 Btu/hr of heat into the tank.

- The COP was reduced to 60% - 75% of the estimated COP as a result of the decreased heating capacity. The observed power draw of the HPWH compressor remains consistent with the power estimated from the manufacturer's compressor map. Because we are providing less heating capacity with the same amount of energy, the efficiency is reduced by a factor similar to the reduction in heating capacity.
- The HPWH seemed to perform best when the superheat was adjusted so the compressor shell was between 80°F & 120°F. When the compressor shell temperature fell below 80ºF (because of liquid returning to the compressor), the average power draw of the compressor tended to be higher than normal. When the shell temperature exceeded 120ºF, the effectiveness of the evaporator was decreased (more of the evaporator was required to provide superheat).

The detrimental effect of these conditions on the HPWH are observed in the state-point diagram presented in Figure 1-3.
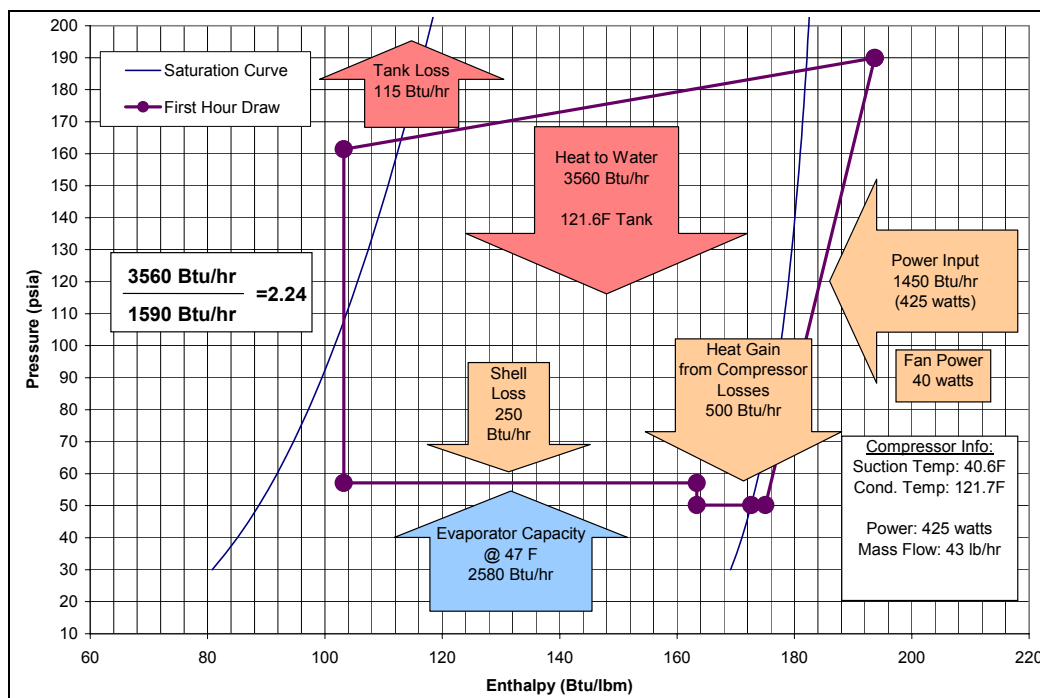
Figure 1-3: System State Points During Initial Testing

In addition, there was insufficient refrigerant flow through the thermostat expansion valve (TXV). There were times when the external overload protection relay would prevent the compressor from starting after it had cycled off. This always occurred when we tried to start the compressor after a short off-cycle period (5 minutes or more). We found that upon shut-down the TXV prevented equalization of high-side and low-side refrigerant pressures. Once the compressor was running, it never tripped the overload.

We addressed the above deficiencies through a series of component modifications, including:

- Replacing the 5/16-in. suction line with a larger, 3/8-in. suction line. This greatly decreased the suction line pressure drop (to less than 1.0 psi).

- Increasing the face area of the evaporator. This modification actually simplified the design of the shroud covering the heat-pump components by extending the evaporator sufficiently close to the shroud so that a seal can be formed between the evaporator and shroud, preventing air

recirculation through the evaporator. Furthermore, the increased face area raises the evaporating temperature, because the capacity is essentially fixed. Thus, the evaporator becomes more sensible and reduces the amount of condensate generated. This also leads to improved efficiency of the compressor.

- The compressor was completely insulated with approximately 3/8-in of Armorflex™ insulation. By insulating the compressor shell, the excess heat generated by the compression process directly heats the discharge gas (rather than heating the incoming liquid refrigerant, or being used to heat the air that passes through the evaporator, thus indirectly increasing the evaporating capacity). The additional heat gained by the discharge gas directly increases the heating capacity of the HPWH. One potential drawback to insulating the compressor is an increased risk of a high discharge temperature, and subsequent reduction in the life expectancy of the compressor. The compressor manufacturer suggests that discharge temperature should not exceed 240ºF.

- The TXV was specified with a balance-port to provide pressure equalization between cycles of the HPWH. Given the relatively low capacity of the refrigeration system, use of the TXV having a bleed port still requires up to two or three minutes for pressures to equalize. However, the micro-controller can lock out the compressor for this time period with no meaningful impact on HPWH performance.

After incorporating the above design refinements, the HPWH system was tested with positive results. These results are described in Section 2 of this attachment.

2.0   Prototype HPWH Laboratory Testing

We completed various laboratory tests in order to verify that the performance of the HPWH satisfied the design requirements. We selected the tests to exercise the HPWH in a broad range of possible conditions and installations. The HPWH testing included the following general tests:

- Heat-up test. This test involves heating a cold tank of water from about 57°F to about 150°F (in order to develop a performance map for the HPWH);

- DOE Energy Factor Test[2];

- Closet Tests. These tests consist of placing the HPWH within a utility closet[3] and, starting with a warm tank (at about 135°F), performing an extended water draw (24 gallons) over an 8-minute period. These tests provide performance data for both the HPWH and the CMS in what is expected to be a typical installation of the HPWH; and

- Garage Tests. These tests involve exposing the HPWH to the ambient conditions expected within a typical household garage installation. The test is initiated with an extended water draw (24 gallons in 8-minutes) from an initially warm tank of water (135°F).

The HPWH tests performed are presented in Table 2-1.

Table 2-1: Performance Testing Matrix

Test No.	Type	Installation	Envirnm. Condition	Condensate Disposal	Test Procedure
1	Heat-Up	--	70°F/50% RH	Drain[a]	Heat-Up
2	Energy Factor	--	67.5°F/50% RH	Drain	DOE Energy Factor
3	24 gal. Draw	Closet	70°F/50% RH	Drain	Closet Testing
4	24 gal. Draw	Closet	80°F/50% RH	Drain	Closet Testing
5	24 gal. Draw	Garage	90°F/80% RH	Drain	Garage Testing
6	24 gal. Draw	Garage	90°F/65% RH	Drain	Garage Testing
7	24 gal. Draw	Garage	50°F/80% RH	Drain	Garage Testing

a) Condensate was drained from the evaporator drain pan and weighed.

2.1. Heat-Up Test

The results for the heat-up test using the third-generation HPWH are presented in Table 2-2. Also presented in the table are the results from previous-testing of our second-generation HPWH. Appendix A, Part 1, lists the test procedure for the heat-up test. The graphical results of the heat-up test are presented in Appendix B.

Table 2-2: Results of Heat-Up Test

Parameter	Third-Generation CEC Prototype	2nd-Generation Prototype	% Change
Avg. Tank Temperature Range	57°F – 151°F	64°F – 151°F	N/A
Sat. Suction Temperature at 120°F	40°F	50°F	N/A
Sat. Discharge Temperature at 120°F	130°F	125°F	N/A
Maximum Discharge Temperature	230°F	178°F	N/A
Range of Superheat	4°F – 16°F	4°F – 28°F	N/A
Range of Subcool	15°F – 45°F	9°F – 24°F	N/A
Power Input at 120°F	435 Watts	375 Watts	16%
Heating Capacity at 120°F	3500 Btu/hr	2700 Btu/hr	30%
COP at 120°F	2.2	2.1	5%

The third-generation HPWH compares favorably to the second-generation unit in many aspects. Most notably, we have increased the efficiency by 5% while also increasing the heating capacity by 30%. In fact, the heating capacity remains greater than 3000 Btu/hr while the COP remains above 1.5 for the entire range of temperatures. Potentially, this has the effect of reducing any hot-water runouts (with no additional loss of efficiency) that may have been experienced otherwise.

The discharge temperature approaches, but never exceeds, 230ºF (even with the average tank temperature at 150ºF) and the compressor shell temperature remains below 180ºF. The sub-cooling at the condenser outlet ranges from 15ºF to 45ºF. Although there is a +/- 7ºF variation in the evaporator superheat, it averages 10ºF throughout the tests. Interestingly, the saturated condensing temperature is exactly 10°F greater than the average tank temperature, precisely what we predicted during our design effort.

2.2. DOE Energy Factor Test

The DOE Energy Factor test was performed at Oak Ridge National Laboratory (ORNL)[4]. A DOE 24-Hour Simulated Use Test (Energy Factor) was completed per the procedure presented in Appendix C, along with detailed charts of results. A comparison of the second-generation and third-generation results are provided in Table 2-3.

Table 2-3: Results of DOE Energy Factor Test

	Third-Generation CEC Prototype	2nd-Generation Prototype	%Change
Energy Value of Heated Water	41,3000 Btu/hr	41,300 Btu/hr	0%
HP Power (compressor fan)	14,860 Btu/hr	13,540 Btu/hr	10%
Element Power (upper)	0	3,660 Btu/hr	-100%
Modified Energy Consumption[a]	16,874 Btu/hr	21,480 Btu/hr	-21%
Recovery Efficiency[b]	3.7	2.2	68%
Energy Factor[c]	2.48	1.92	29%

a) Effectively the total energy consumed by the HPWH during the 24 hrs. Defined in 10-CFR-430, Subpart B, Appendix E, Section 6.1.6.
b) Describes the efficiency of the HP"WH during the first hour water draw. Defined in 10-CFR-4230, Subpart B, Appendix E, Section 6.1.6
c) Indicative of in-situ performance, Defined in 10-CFR-430, Subpart B, Appendix E, Section 6.1.6.

The calculated Energy Factor of the Third-Generation Prototype was a remarkable 2.48. This represents a significant improvement of nearly 30% over the second-generation HPWH. Furthermore, this exceeds our design goal of 2.0 by nearly 25%.

Two major factors responsible for the improvements are:

- The HPWH had enough heating capacity during the initial hours of the test to prevent the upper element from energizing during the sixth draw, and

- The recovery efficiency during the first hour was appreciably higher than was realized in the second-generation prototype (by nearly 70%).

One might ask how it is possible to approach the EF of currently available HPWHs when those products use compressors more efficient (and more expensive) than our design. A possible explanation is that the water pump required by the Crispaire/Etech HPWH lowers the efficiency, offsetting the advantage it would otherwise have. However, we currently have no hypothesis as to why the EF of the DEC Therma-Stor HPWH does not far exceed the EF we have achieved (although, at an EF of 2.6, it's performance is somewhat higher).

2.3. Simulation Tests

The results from the simulation tests show a marked improvement in performance of the third-generation HPWH over the second-generation unit. The sections below detail the results of both the closet tests and the garage tests.

It should be noted, however, that results of these some of these tests might be inaccurate and not represent the true performance of the HPWH. A slow refrigerant leak was recently discovered in the one of the shrader valves used to connect the pressure transducers. This leak may have affected the following tests:

- 90°F/65% RH Garage Test;
- 70°F/50% RH Closet Test; and
- 90°F/65% RH Closet Test.

These tests exhibit higher-than-normal superheat values, lower capacities, and lower efficiencies, which are all indicative of low-charge in the system (due to the leak). However, the results still provide some insight into the system performance and for that reason are included in the comparisons below.

2.3.1. Garage Tests

The results of the three garage tests are presented in Tables 2-4, 2-5, and 2-6. The data from the garage tests are presented in Appendix D.

It is encouraging to see a significant increase in heating capacity all three tests. COP is higher for two of the three tests. Only the COP during the 90°F/65% RH test is lower, but that may be attributed to the leak described above.

The discharge temperatures are higher than previously experienced with the second-generation unit for two of the three tests. However, the refrigerant system temperatures are still within the limits set forth in the design effort.

Table 2-4: Summary Results of 50°F/80% RH Garage Test

Parameter	50°F/80% RH Garage		
	Third-Generation CEC Prototype	2nd-Generation Prototype	% Change
Avg. Tank Temperature Range	105°F-145°F	102°F-135°F	N/A
Sat. Suction Temperature at 120°F[a]	30°F	38°F	N/A
Sat. Discharge Temperature at 120°F	125°F	125°F	N/A
Maximum Discharge Temperature	218°F	150°F	N/A
Range of Superheat	4°F-16°F	3°F	N/A
Range of Subcool	20°F-60°F	10°F-15°F	N/A
Generated Condensate[b]	33.4 oz.	19.2 oz.	74%
Power Input at 120°F	420 Watts	350 Watts	20%
Heating Capacity at 120°F	3000 Btu/hr	1700 Btu/hr	76%
COP at 120°F	2.1	1.4	50%

a) Average tank temperature of 120°F
b) Total amount of condensate generated during the test

Table 2-5: Results of 90°F/80% RH Garage Test

Parameter	90°F/80% RH Garage[a]		
	Third-Generation CEC Prototype	2nd-Generation Prototype	% Change
Avg. Tank Temperature Range	106°-143°F	100°F-135°F	N/A
Sat. Suction Temperature at 120°F[b]	60°F	60°F	N/A
Sat. Discharge Temperature at 120°F	131°F	125°F	N/A
Maximum Discharge Temperature	225°F	185°F	N/A
Range of Superheat	4°F-16°F	5°F-30°F	N/A
Range of Subcool	10°F-45°F	12°F-28°F	N/A
Generated Condensate[c]	–	24 oz.	–
Power Input at 120°F	585 Watts	400 Watts	46%
Heating Capacity at 120°F	7000 Btu/hr	4400 Btu/hr	59%
COP at 120°F	3.5	3.2	9.4%

a) Second-generation conditions were 80°F/80%RH

b) Average tank temperature of 120°F

c) Total amount of condensate generated during the test

Table 2-6: Results of 90°F/65% RH Garage Test

Parameter	90°F/65% RH Garage[a]		
	Third-Generation CEC Prototype	2nd-Generation Prototype[b]	% Change
Avg. Tank Temperature Range	105°F-140°F	102°-135°F	N/A
Sat. Suction Temperature at 120°F[c]	60°F	65°F	N/A
Sat. Discharge Temperature at 120°F	130°F	125°F	N/A
Maximum Discharge Temperature	170°F	190°F	N/A
Range of Superheat	15°F-25°F	10°F-50°F	N/A
Range of Subcool	0°F-45°F	20°F-30°F	N/A
Generated Condensate[d]	34.0 oz.	20.8 oz.	63%
Power Input at 120°F	580 Watts	420 Watts	38%
Heating Capacity at 120°F	5500 Btu/hr	5000 Btu/hr	10%
COP at 120°F	2.7	3.5	-23%

a) Second-generation conditions were 100°F/50%RH

b) Test results exhibit signs of low charge due to leakage from instrumentation

c) Average tank temperature of 120°F

d) Total amount of condensate generated during the test

2.3.2. Closet Tests

In the closet, the performance of the HPWH was satisfactory. The results of the testing are presented in Table 2-7. The graphical data for the closet tests are included in Appendix E.

The heating capacity was lower than the heat-up and garage tests but was still acceptable at 2300 Btu/hr. The COP is similar to that seen in the second-generation HPWH.

The closet temperature remains above 44°F throughout the tests, compared to 52°F for the second-generation prototype. Relative humidity remained below 67 percent, compared to 80

percent for the second-generation prototype. There was no condensate observed on the either the interior (after the test) or exterior walls. The relatively low closet temperature (44°F) suggests that we are at the threshold limit for the heating capacity and do not want to increase it any further. Although it may seem counterintuitive, more heating capacity is not always better. Higher heating capacities require greater heat-transfer rates from the air to the evaporator. In small enclosures (such as utility closets) excessive capacity can lower the closet-air temperature to a point where either condensation begins to form on closet walls or the evaporator begins to frost, requiring the heat pump to be shut off.

Table 2-7: Results of Closet Tests

Parameter	70°F/50% RH Outside Closet		
	Third-Generation Prototype	2nd-Generation Prototype[a]	% Change
Avg. Tank Temperature Range	105°F-130°F	102°F-135°F	N/A
Minimum Closet Temperature	44°F	52°F	N/A
Maximum Closet RH	67%	80%	-16%
Sat. Suction Temperature at 120°F[b]	30°F	40°F	N/A
Sat. Discharge Temperature at 120°F	124°F	138°F	N/A
Maximum Discharge Temperature	212°F	155°F	N/A
Range of Superheat	27°F-40°F	3°F	N/A
Range of Subcool	44°F-48°F	13°F	N/A
Generated Condensate[c]	4 oz.	6 oz.	-33%
Power Input at 120°F	400 Watts	340 Watts	18%
Heating Capacity at 120°F	2300 Btu/hr	2200 Btu/hr	5%
COP at 120°F	1.7	1.9	-11%

a) Average of two tests using condensate drains and in an unsealed closet. See [ADL 1998; Figures 6.7 to 6.12].

b) Average tank temperature of 120°F

c) Total amount of condensate generated during the test.

ENDNOTES

[1] A potentiometer modified by connecting the middle terminal to either outside terminal.

[2] Department of Energy, *10-CFR-430, Subpart B, Appendix E*, Section 5.1.5 – 24 Hour Simulated Use Test

[3] A utility closet with interior dimensions of 8-ft. x 30-in. x 30-in. with a 24-in x 80-in. door. The walls are a stud construction, consisting of 2x4s and drywall. Both the floor and ceiling were constructed using 3/8"

[4] ORNL has served as an integral member of the HPWH development team for a number of years.

Appendix A - Test Procedures

Testing Procedures for HPWH Heat-Up Tests

1. Set the environmental chamber to 70°F / 50% RH climate. Verify the test conditions are within +/-2 ºF of the desired temperature and +/- 2.5% of the desired RH.

2. Make certain power is disconnected to the HPWH.

3. Start the test with a cold tank of water. This is accomplished by either filling the water tank, or performing an extended draw (at roughly 3 gpm for at least 20 minutes).

4. Disable the upper element (disconnect wiring).

5. Start the data acquisition system and begin recording data.

   - Verify the operation of the thermocouples and pressure gages. Make certain all thermocouple beads on the refrigeration system are insulated from the ambient air.
   - Verify the operation of the power supply.
   - Verify that the evaporator pan is dry. Make certain the drain hose is connected to the evaporator pan and is free and clear of debris. Make certain there are no restrictions in the hose. Make certain the condensate collection pan is dry.
   - Verify the RH sensor and ambient thermocouples are located in the inlet air-stream of the HPWH (outside of the shroud – in front and center of the inlet air grille).

6. Turn on the HPWH and allow the water tank to reach 140°F (+/- 5°F).

7. Stop the data acquisition system and record observations in a laboratory note book.

Testing Procedures for HPWH Performance Tests

The test procedure described below is for testing the HPWH with the conditions presented in below.

Test No.	Type	Installation	Envirnm. Condition	Condensate Disposal	Test Procedure
1	24 gal. Draw	Closet	70F/50%	Drain	Closet Testing
2	24 gal. Draw	Closet	80F/50%	Drain	Closet Testing
3	24 gal. Draw	Garage	90F/80%	Drain	Garage Testing
3	24 gal. Draw	Garage	90F/65%	Drain	Garage Testing
4	24 gal. Draw	Garage	50F/80%	Drain	Garage Testing

Test Procedure

1. Set the environmental chamber to specified climate. Verify the test conditions are within +/-2 ºF of the desired temperature and +/- 2.5% of the desired RH.

2. Start the data acquisition system and begin recording data.

3. Draw water from the tank until the HPWH compressor starts and then stop the draw.
   - Verify the supply water is 58ºF (+/- 2ºF). Verify the flow rate is 3 gpm (+/- 0.25 gpm)
   - Verify the operation of the thermocouples and pressure gages. Make certain all thermocouple beads on the refrigeration system are insulated from the ambient air.
   - Verify the operation of the power supply.
   - Verify that the evaporator pan is dry. Make certain the drain hose is connected to the evaporator pan and is free and clear of debris. Make certain there are no restrictions in the hose. Make certain the condensate collection pan is dry.
   - Verify the RH sensor and ambient thermocouples are located in the inlet air-stream of the HPWH (outside of the shroud – in front and center of the inlet air grille).
   - Verify the thermostat cutout occurs when the average tank temperature reaches 135ºF (+5ºF/-0ºF) and the HPWH shuts off.

4. If performing a closet test, shut the closet door prior to Step 5.

5. Take a water draw of 24 gallons. This is achieved by taking an 8-minute draw, with the flow meter set to 3 gpm (+/- 0.25 gpm). Allow the HPWH to again reach the set-point temperature and the heat pump compressor to shut-off.

6. Stop the data acquisition system and record observations in a laboratory note book. In the case of the closet test, look for the presence of moisture on the interior or exterior closet walls.

7. Drain any remaining condensate from the evaporator pan into the condensate collection pan. Weigh and record the amount of condensate in the collection pan.

Appendix B - Heat-Up Test
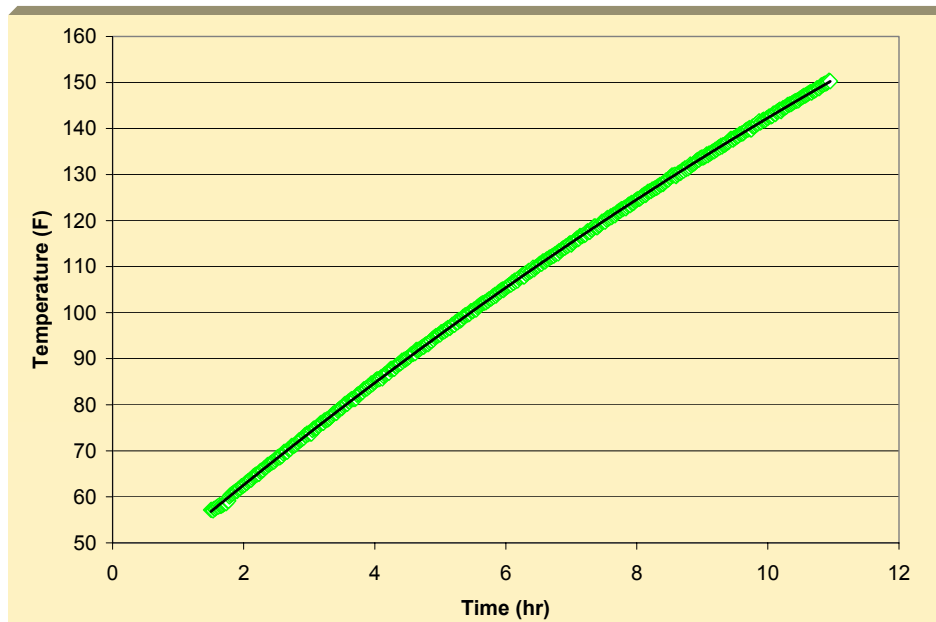
Figure B-1 Average Tank Temperature for heat-up Test[1]
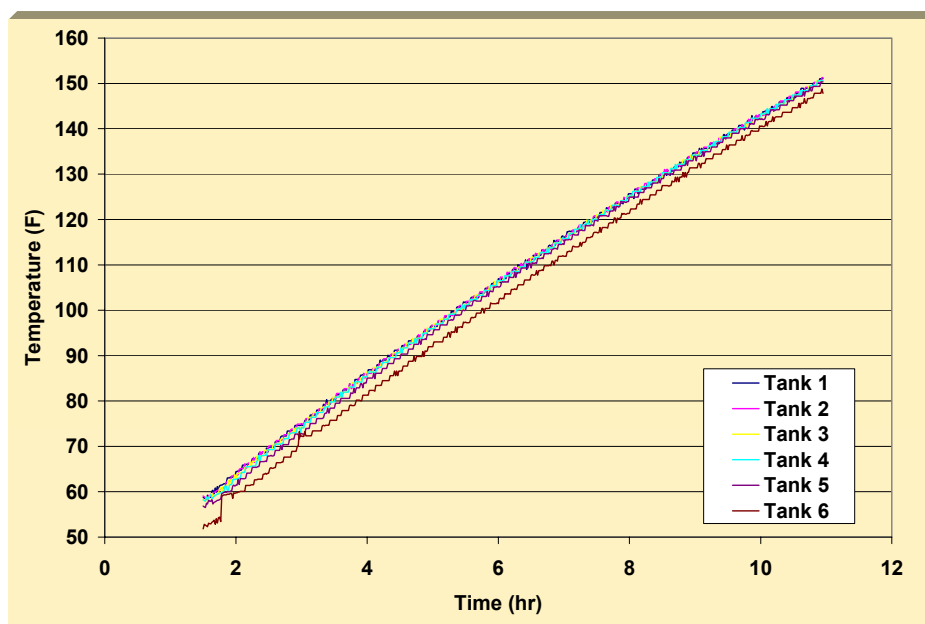
Figure B-2 Tank Temperatures for Heat-Up Test

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[1] The solid line represents the results of the linear regression performed on the average tank temperature

Figure B-2 Tank Temperatures for Heat-up Test
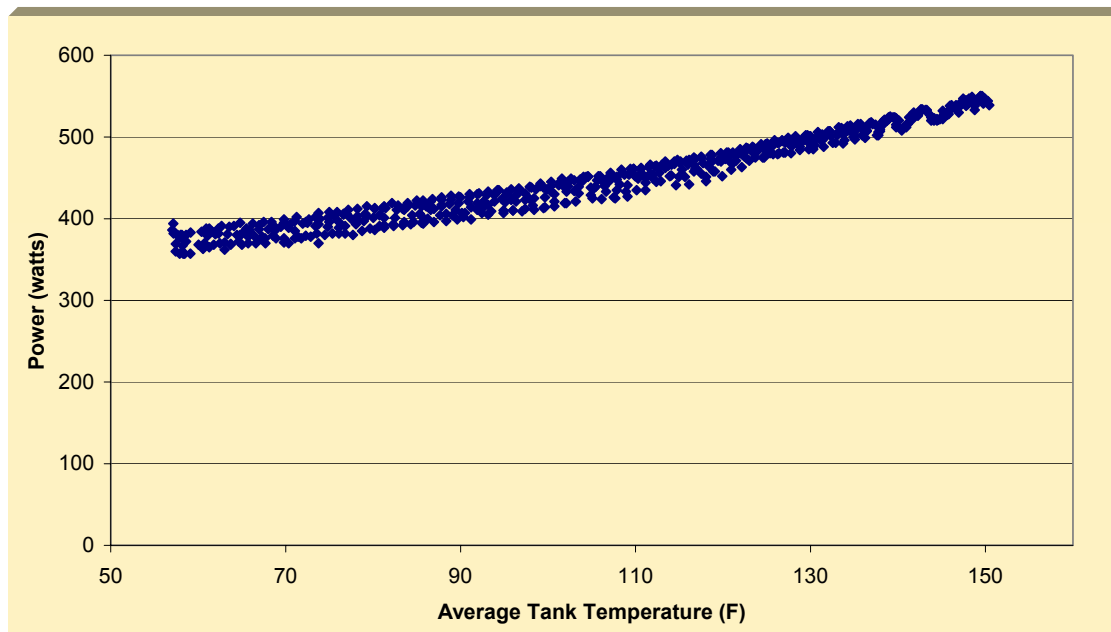
Figure B-3 HPWH Power for Heat-up Test
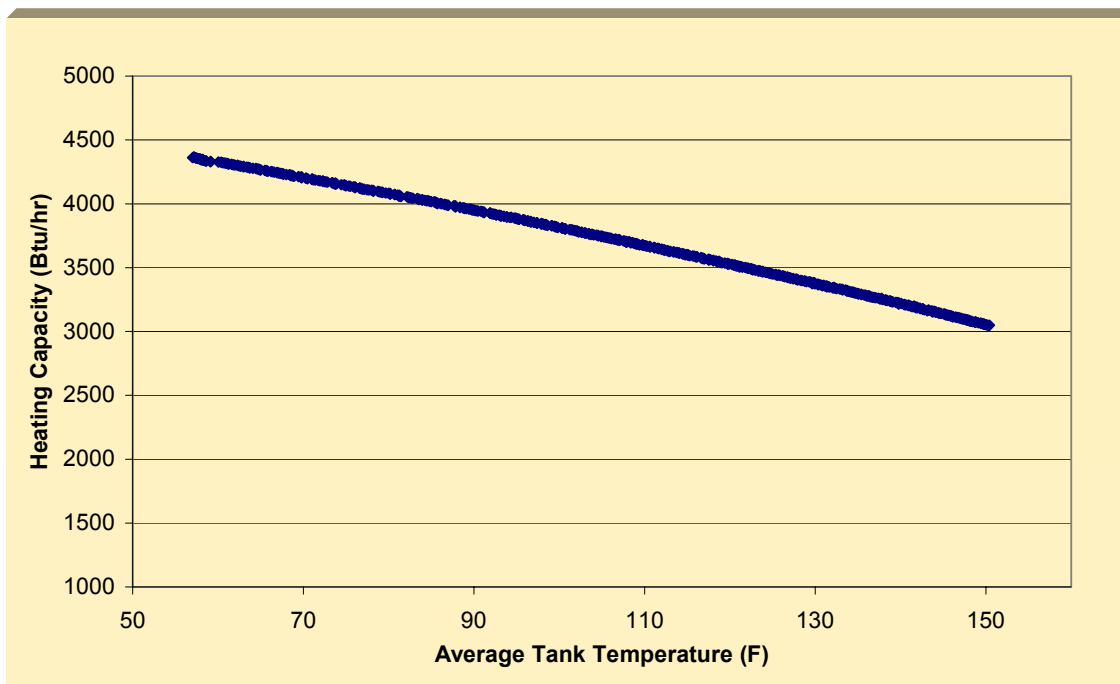
Figure B-4 HPWH Capacity for for Heat-up Test[2]

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[2] Calculated from linear regression of the average tank temperature.
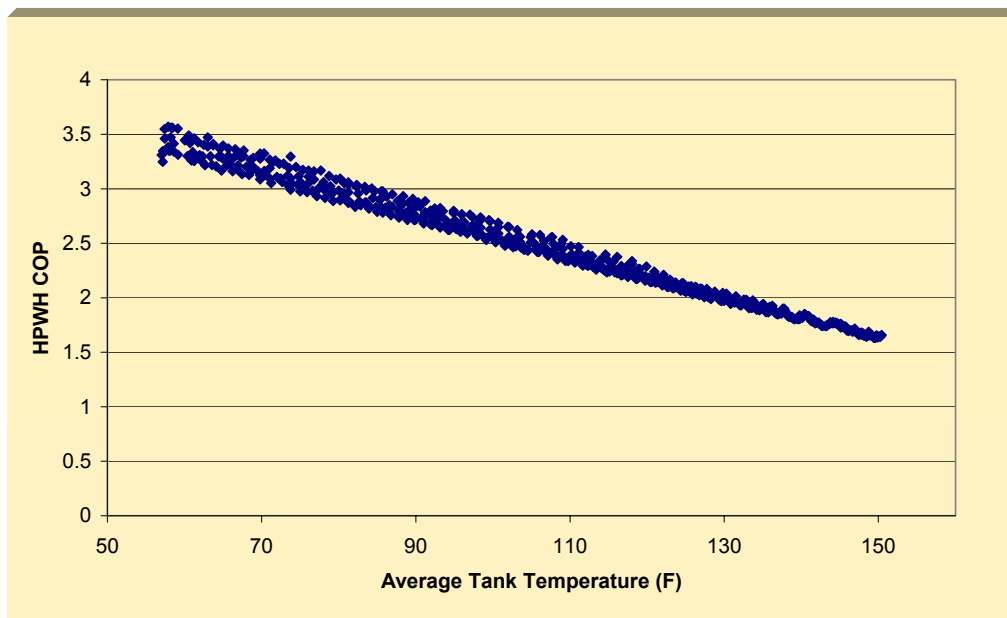
Figure B-5 HPWH COP for Heat-up Test[3]
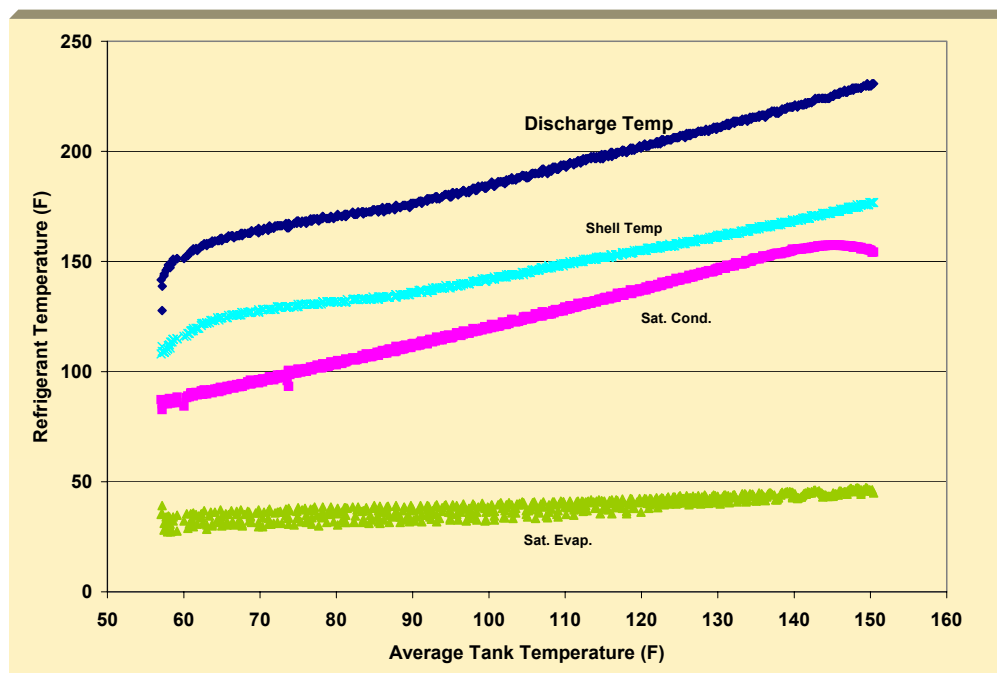
Figure B-6 Refrigerant System Temperatures for Heat-up Test

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[3] Calculated using raw power data and the smoothed heating capacity data.
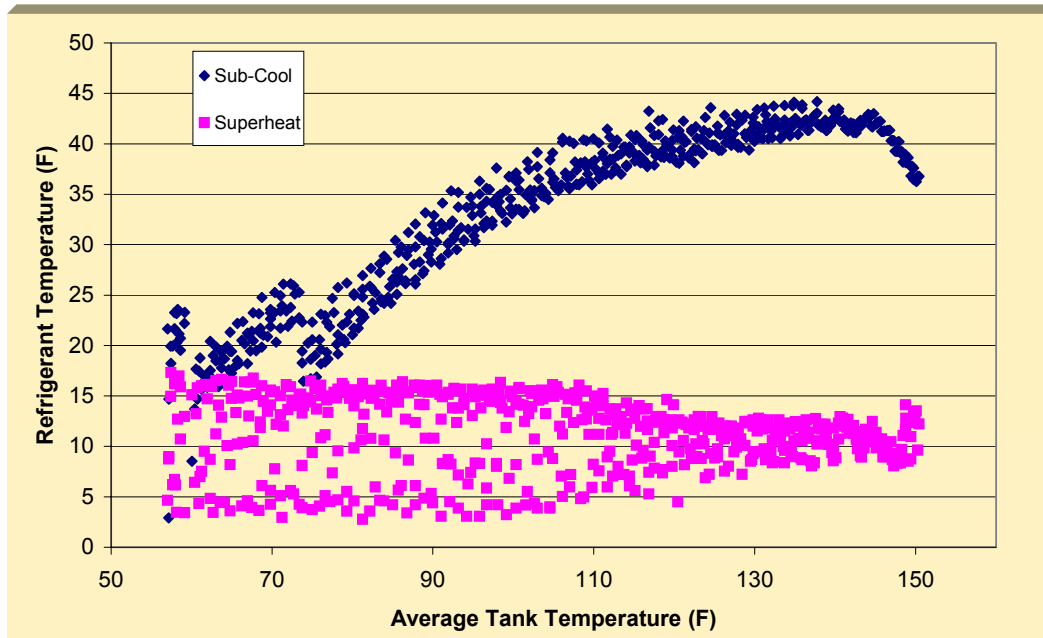
Figure B-7 Superheat and Sub-Cool for Heat-up Test
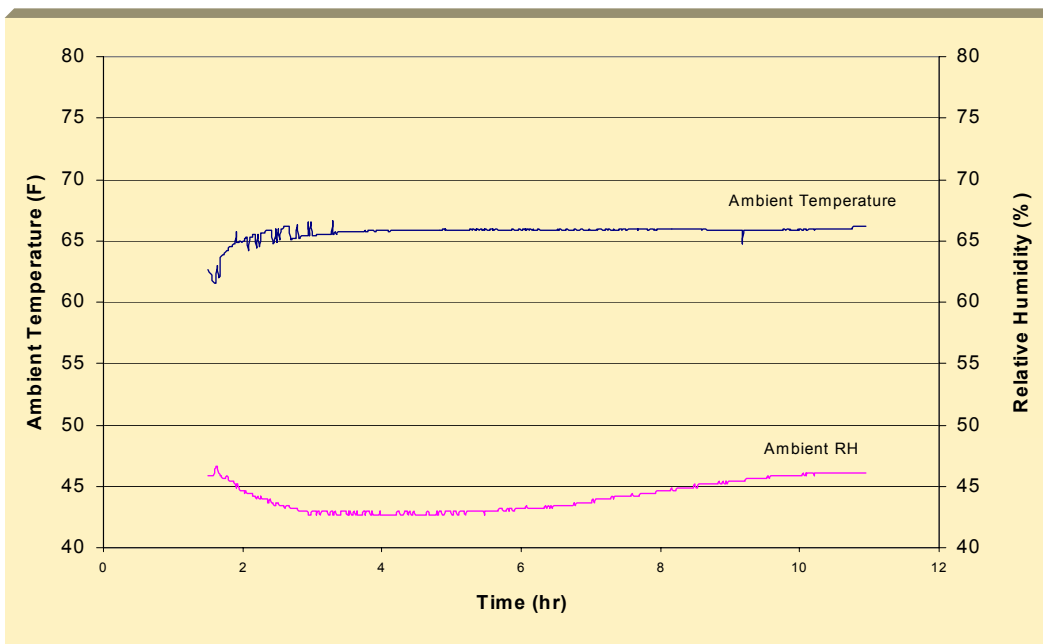
Figure B-8 Ambient Conditions during Heat-up Test

Appendix C - Energy Factor Tests

Problem Statement: Determine the Energy Usage Factor as directed in 10-CFR-430, Subpart B, Appendix E, Section 6.1.7, *Energy Factor*

References: [1] 10-CFR-430, Subpart B, Appendix E
[2] ORNL Test Results of March 24, 2000

Definitions: $F \equiv R$

Solution:

We follow [1] from 6.1.3 to 6.1.7 to determine the Energy Factor

6.1.3 Recovery Efficiency

$M_1 := 10.717 \cdot gal \cdot 8.333 \cdot \dfrac{lb}{gal}$ $M_1 = 89.3048 \cdot lb$ Mass of water withdrawn during the first draw

$T_{del_1} := 134.4 \cdot F$ Average delivery temperature for the first draw

$T_{in_1} := 58.8 \cdot F$ Average inlet temperature for the first draw

$Cp_1 := 1.0 \cdot \dfrac{BTU}{lb \cdot F}$ Specific heat of water at the average water temperature

$V_{st} := 45 \cdot gal$ Storage tank capacity (approximate)

$T_{max_1} := 132.6 \cdot F$ Maximum mean tank temperature recorded after cutout following the first draw[1]

$T_o := 134.7 \cdot F$ Maximum mean tank temperature recorded prior to the first draw

$\rho := 62.4 \cdot \dfrac{lb}{ft^3}$ Density of water at the average temperature

$Cp_2 := 1.0 \cdot \dfrac{BTU}{lb \cdot F}$ Specific heat of water at the average water temperature

$Q_r := 1628.282 \cdot BTU$ Measured total energy used by the water heater between cutout prior to the first draw and cutout following the first draw[2]

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[1] water heater never achieved cutout after first draw, this temperature represents maximum temperature prior to second draw

[2] Total energy between start of test and prior to second draw. See footnote 1.

The recovery efficiency for the water is computed as:

$$\eta_r := \frac{M_1 \cdot Cp_1 \cdot \left(T_{del_1} - T_{in_1}\right)}{Q_r} + \frac{V_{st} \cdot \rho \cdot Cp_2 \cdot \left(T_{max_1} - T_o\right)}{Q_r} \qquad \eta_r = 3.6622$$

6.1.4 Hourly Standby Losses

$Q_{stby} := 0 \cdot BTU$

Total energy consumed (and used for heating the water) between the maximum mean tank temperature after the sixth draw and the end of the 24 hour test.

$T_{24} := 131.5 \cdot F$

Average tank temperature at the conclusion of the 24 hour test.

$T_{su} := 142.5 \cdot F$

Maximum mean tank temperature observed after the sixth draw

$M := V_{st} \cdot \rho \qquad M = 375.375 \cdot lb$

Mass of the water within the storage tank

$Cp := 1.0 \cdot \dfrac{BTU}{lb \cdot F}$

Specific heat of water at the average water temperature

$\tau_{stby} := 14.35 \cdot hr$

Elapsed time between the time at which the maximum tank temperature is observed and the end of the 24 hour test.

The hourly standby losses are computed as:

$$Q_{hr} := \frac{Q_{stby} - \dfrac{M \cdot Cp \cdot \left(T_{24} - T_{su}\right)}{\eta_r}}{\tau_{stby}} \qquad Q_{hr} = 78.5706 \cdot \frac{BTU}{hr}$$

Standby Heat Loss Coefficient

$Tt_{stby} := 136.7 \cdot F$

Average mean tank temperature between the time at which the maximum tank temperature is observed after the sixth draw and the end of the 24 hour test.

$Ta_{stby} := 65.1 \cdot F$

Average ambient temperature between the time at which the maximum tank temperature is observed after the sixth draw and the end of the 24 hour test.

The standby heat loss coefficient for the tank is computed as:

$$UA := \frac{Q_{hr}}{Tt_{stby} - Ta_{stby}} \qquad UA = 1.0974 \cdot \frac{BTU}{hr \cdot F}$$

6.1.5 Daily Water Heating Energy Consumption

$Q := 14866.1 \cdot BTU$ — Total energy used by the water heater during the 24 hour simulated use test.

$T_{24} = 131.5 \cdot F$ — Average tank temperature at the conclusion of the 24 hour test.

$T_o = 134.7 \cdot F$ — Average tank temperature prior to the first draw.

$C_p := 1.0 \cdot \dfrac{BTU}{lb \cdot F}$ — Specific heat of water at the average water temperature

$M = 375.375 \cdot lb$ — Mass of the water within the storage tank

$\eta_r = 3.6622$ — Recovery efficiency, from above.

The daily water heating energy consumption is computed as:

$$Q_d := Q - \frac{M \cdot Cp \cdot (T_{24} - T_o)}{\eta_r} \qquad Q_d = 1.5194 \cdot 10^4 \cdot BTU$$

6.1.6 Adjusted Daily Water Heating Energy Consumption

$Q_d = 1.5194 \cdot 10^4 \cdot BTU$ — Daily water heating energy consumption, from above.

$\tau_{stby2} := 23.633 \cdot hr$ — Total time during the 24 hour test when water was not being withdrawn from the tank.

$T_{stby2} := 129.6 \cdot F$ — Mean tank temperature during the total standby portion, τ_{stby2}, of the 24 hour test.

$Ta_{stby2} := 65.5 \cdot F$ — Average ambient temperature during the total standby portion of the 24 hour test

The adjusted daily water heating energy consumption is computed as:

$$Q_{da} := Q_d - \left[\left(T_{stby2} - Ta_{stby2} \right) - (135 \cdot F - 67.5 \cdot F) \right] \cdot UA \cdot \tau_{stby2}$$

$$Q_{da} = 1.5282 \cdot 10^4 \cdot BTU$$

The energy used to heat water, BTU per day, may be computed as -

$$M := \begin{bmatrix} 10.717 \cdot gal \\ 10.717 \cdot gal \\ 10.717 \cdot gal \\ 10.717 \cdot gal \\ 10.717 \cdot gal \\ 10.717 \cdot gal \end{bmatrix} \cdot 8.343 \cdot \frac{lb}{gal} \qquad T_{del} := \begin{bmatrix} 134.4 \\ 133.6 \\ 130.3 \\ 125.2 \\ 118.2 \\ 112.7 \end{bmatrix} \cdot F \qquad T_{in} := \begin{bmatrix} 58.8 \\ 55.0 \\ 58.8 \\ 58.4 \\ 58.9 \\ 59.5 \end{bmatrix} \cdot F \qquad Cp := 1 \cdot \frac{BTU}{lb \cdot F}$$

$$Q_{HW} := \sum_{i=1}^{6} \frac{M_i \cdot Cp \cdot \left(T_{del_i} - T_{in_i}\right)}{\eta_r} \qquad Q_{HW} = 9.8879 \cdot 10^3 \ \circ BTU$$

The energy required to heat the same quantity of water over a 77F temperture rise, BTU per day, is calculated as -

$$Q_{HW77} := \sum_{i=1}^{6} \frac{M_i \cdot C_p \cdot (135 \cdot F - 58 \cdot F)}{\eta_r} \qquad Q_{HW77} = 1.128 \cdot 10^4 \ \circ BTU$$

The difference between these two values is -

$$Q_{HWD} := Q_{HW77} - Q_{HW} \qquad Q_{HWD} = 1.3916 \cdot 10^3 \ \circ BTU$$

Which must be added to the adjusted daily water heating energy consumption value. Thus, the daily energy consumption value which takes into account that the temperature difference between the storage tank and the ambient temperature may not be 67.5F and the temperature rise across the storage tank may not be 77F is -

$$Q_{dm} := Q_{da} + Q_{HWD} \qquad Q_{dm} = 1.6674 \cdot 10^4 \ \circ BTU$$

6.1.7. Energy Factor

The energy factor is computed as -

$$E_f := \sum_{i=1}^{6} \frac{M_i \cdot Cp \cdot (135.0 F - 58.0 F)}{Q_{dm}} \qquad \boxed{E_f = 2.4774}$$
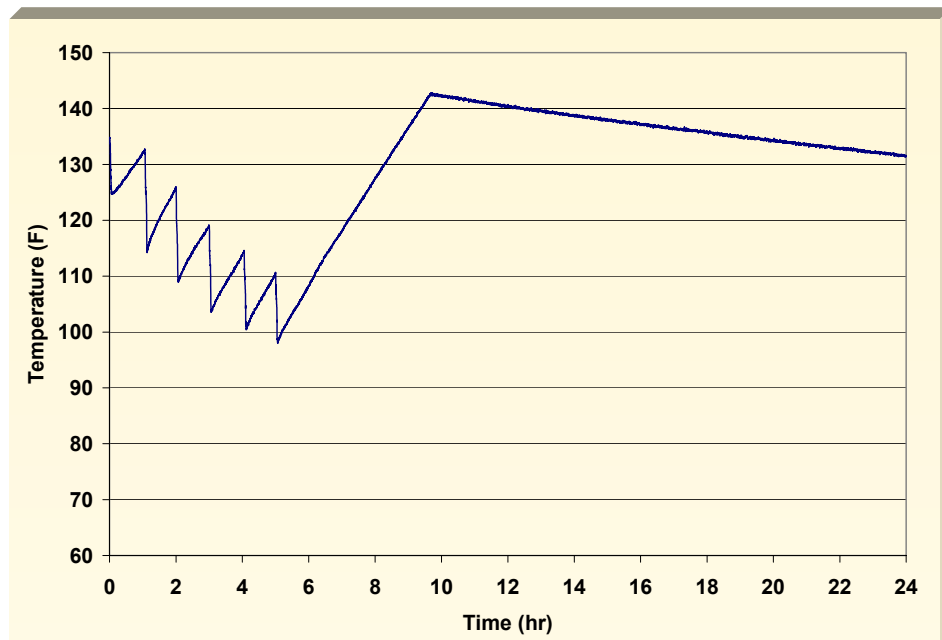
Figure C-1 Average Tank Temperature for DOE Energy Factor Test
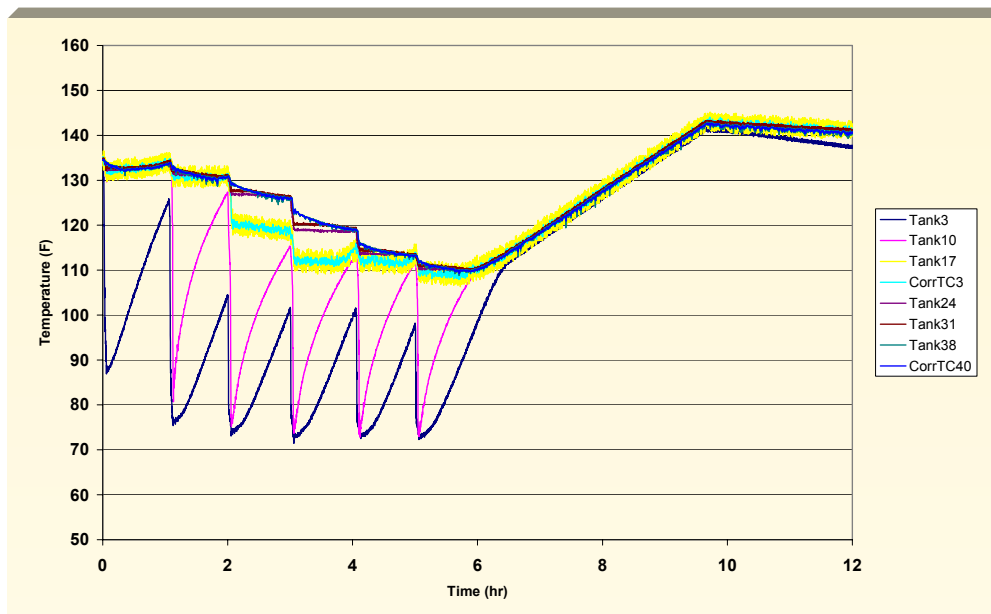
Figure C-2 Tank Temperatures for DOE Energy Factor Test
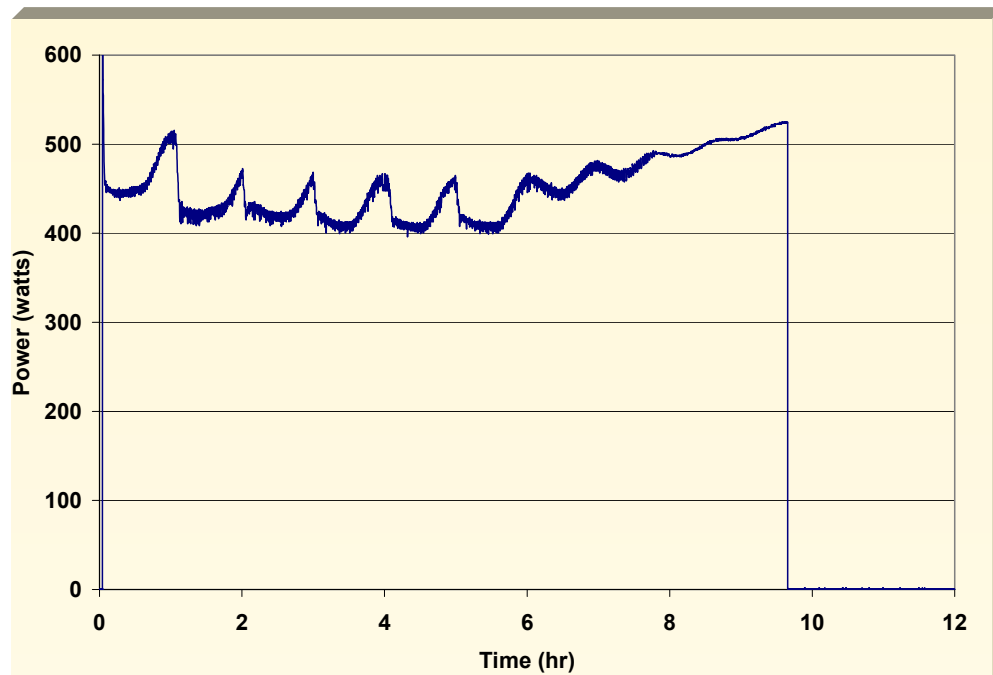
Figure C-3 HPWH Power for DOE Energy Factor Test
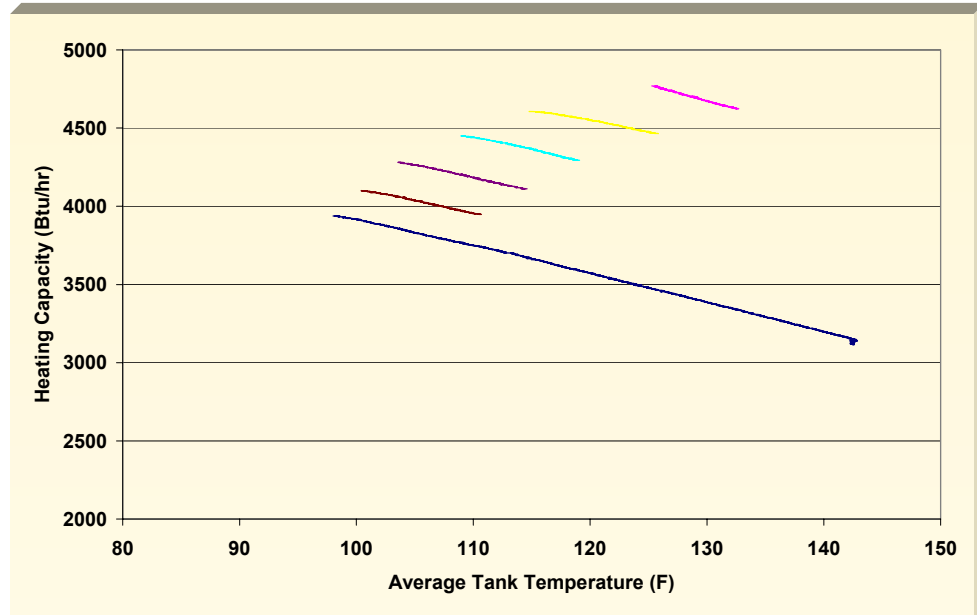
Figure C-4 HPWH Capacity for DOE Energy Factor Test4

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[4] Calculated from linear regression of the average tank temperature.
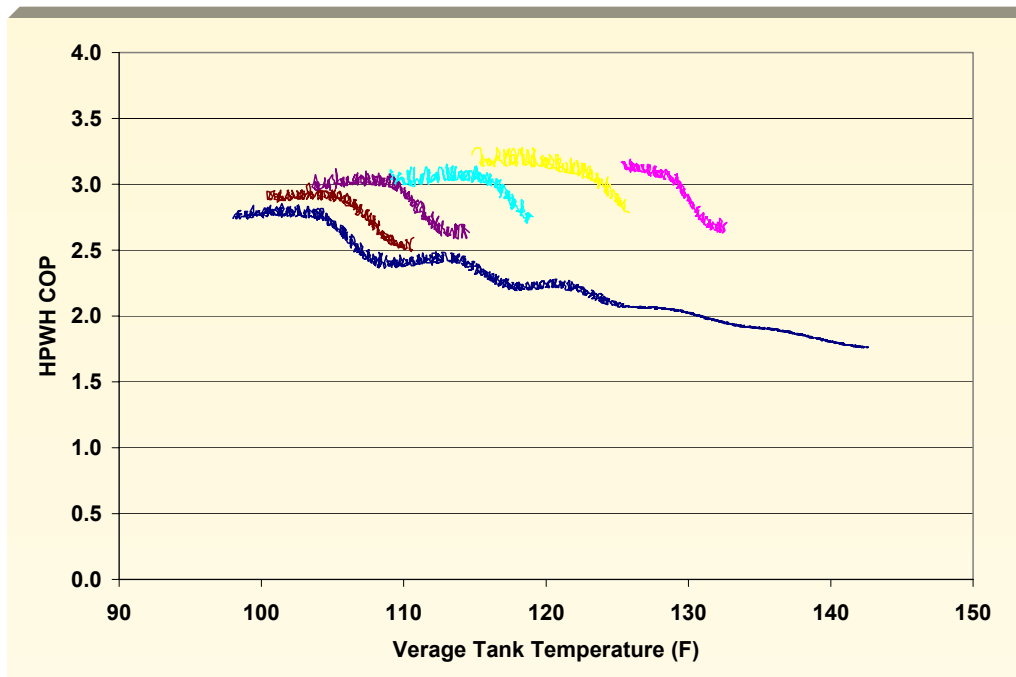
Figure C-5 HPWH COP for DOE Energy Factor Test[5]
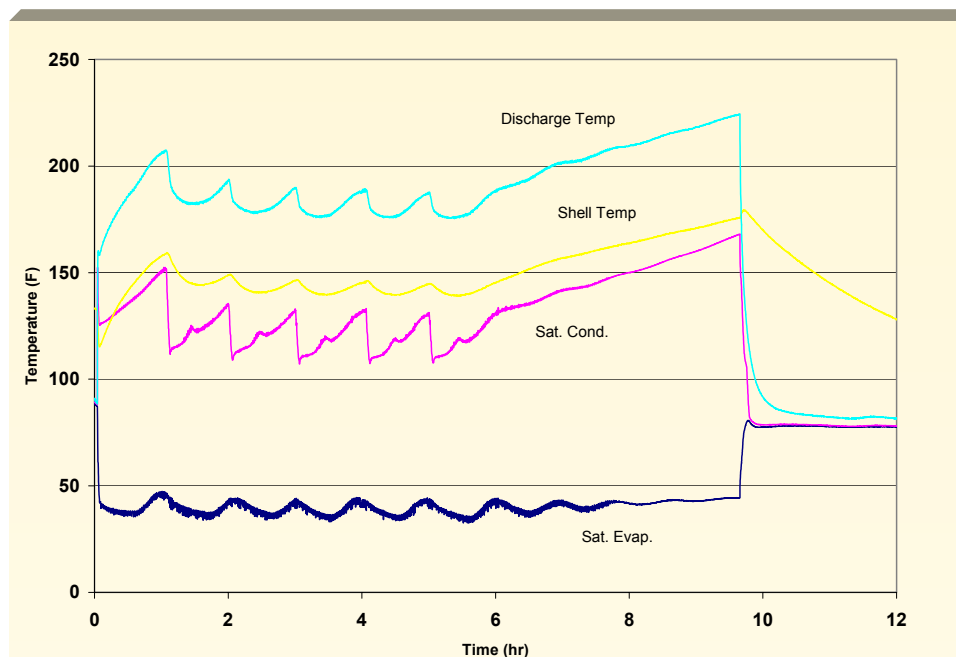
Figure C-6 Refrigerant System Temperatures for DOE Energy Factor Test

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[5] Calculated using raw power data and the smoothed heating capacity data.
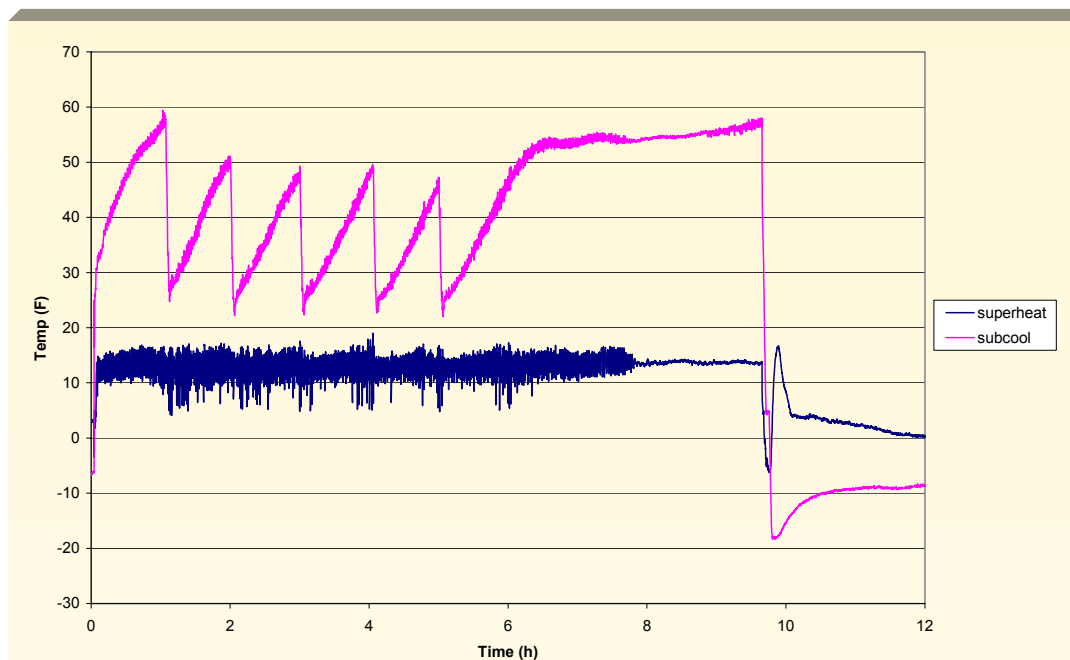
Figure C-7 Superheat and Sub-Cool for DOE Energy Factor Test

Appendix D - Garage Tests
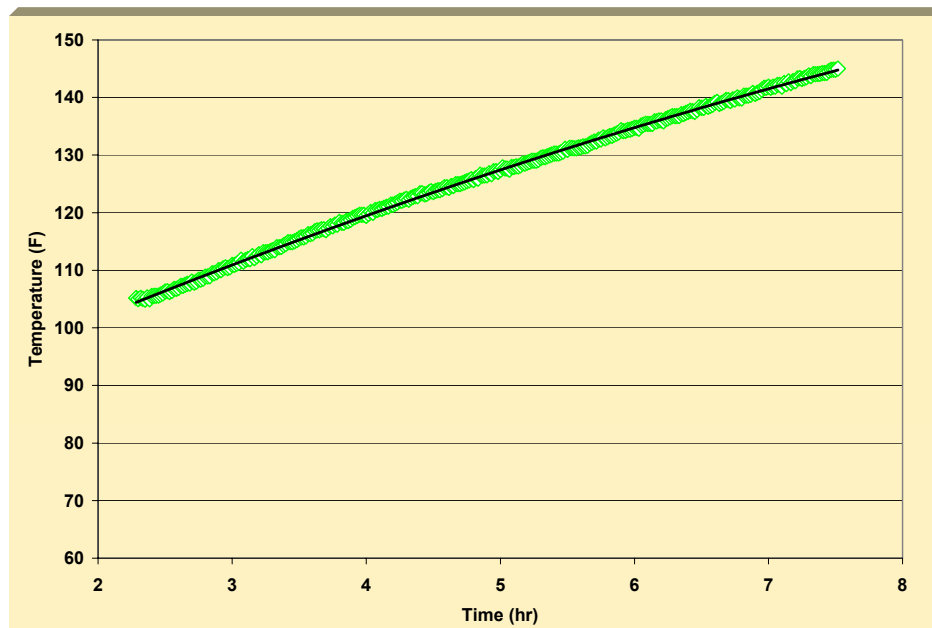
Figure D-1 Average Tank Temperature for 50°F/80% RH Garage Test[6]
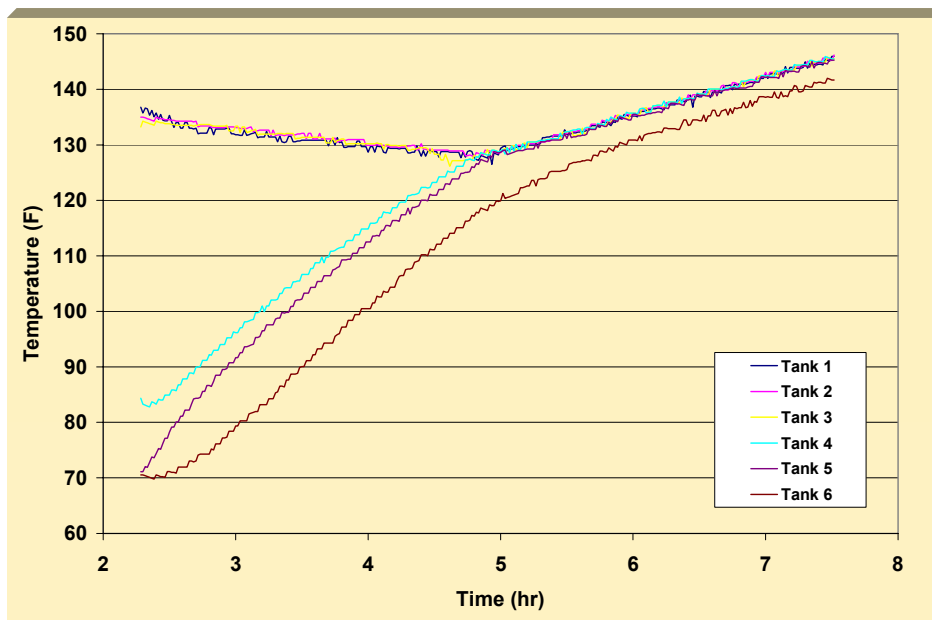
Figure D-2 Tank Temperatures for 50°F/80% RH Garage Test

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[6] The solid line represents the results of the linear regression performed on the average tank temperature
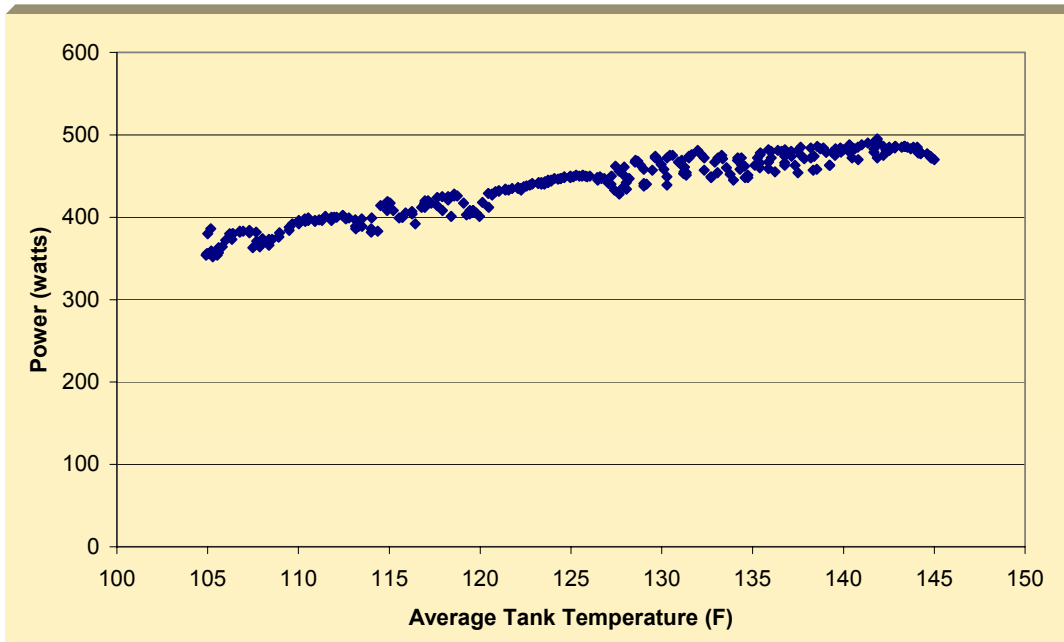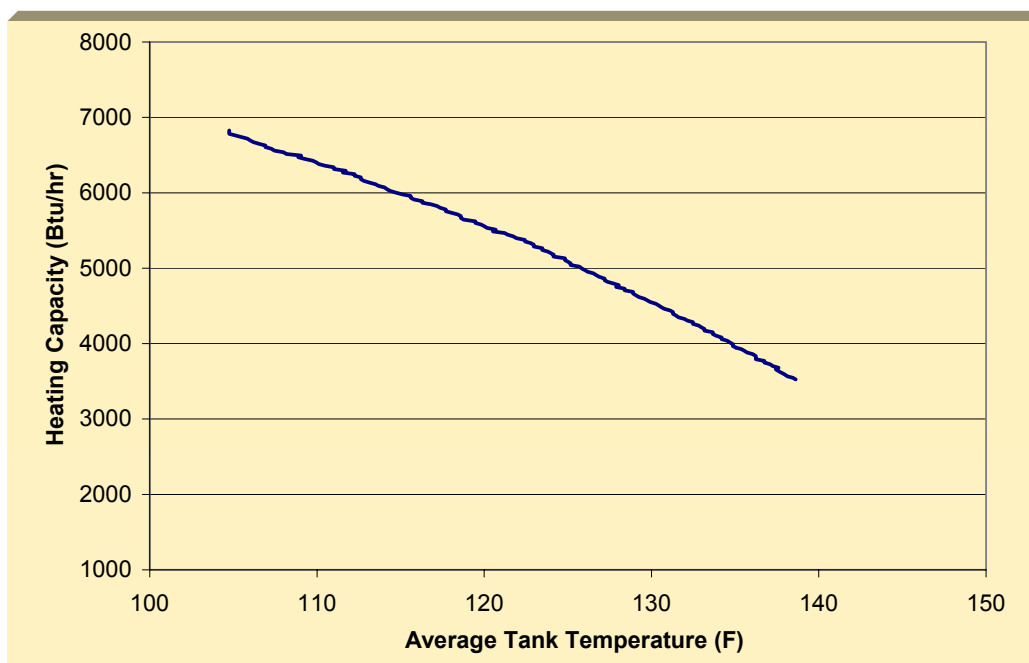
Figure D-3 HPWH Power for 50°F/80% RH Garage Test
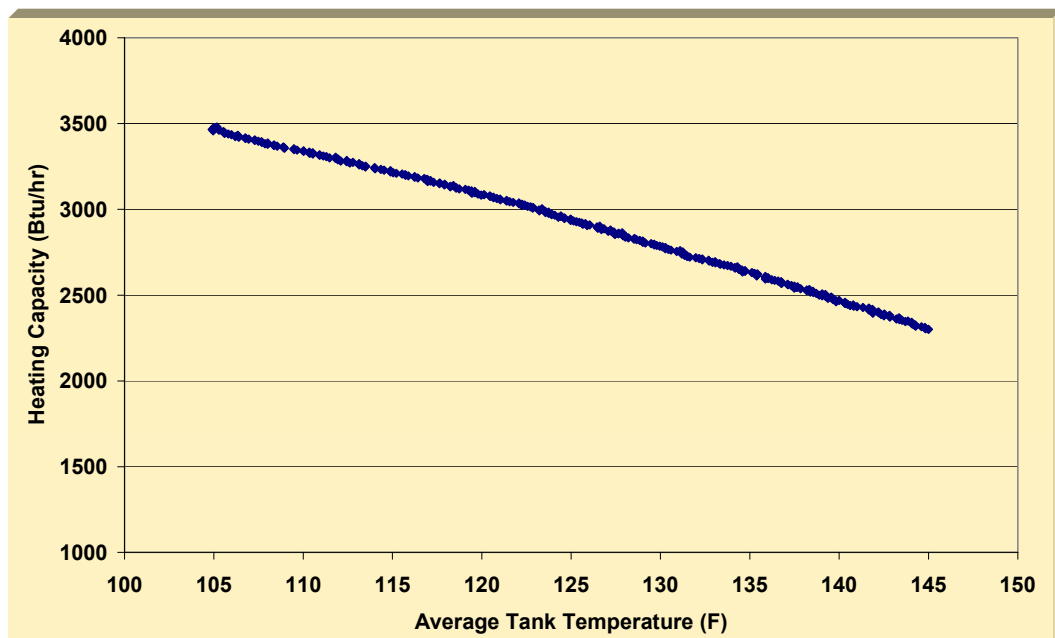
Figure D-4 HPWH Capacity for 50°F/80% RH Garage Test[7]

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[7] Calculated from linear regression of the average tank temperature.
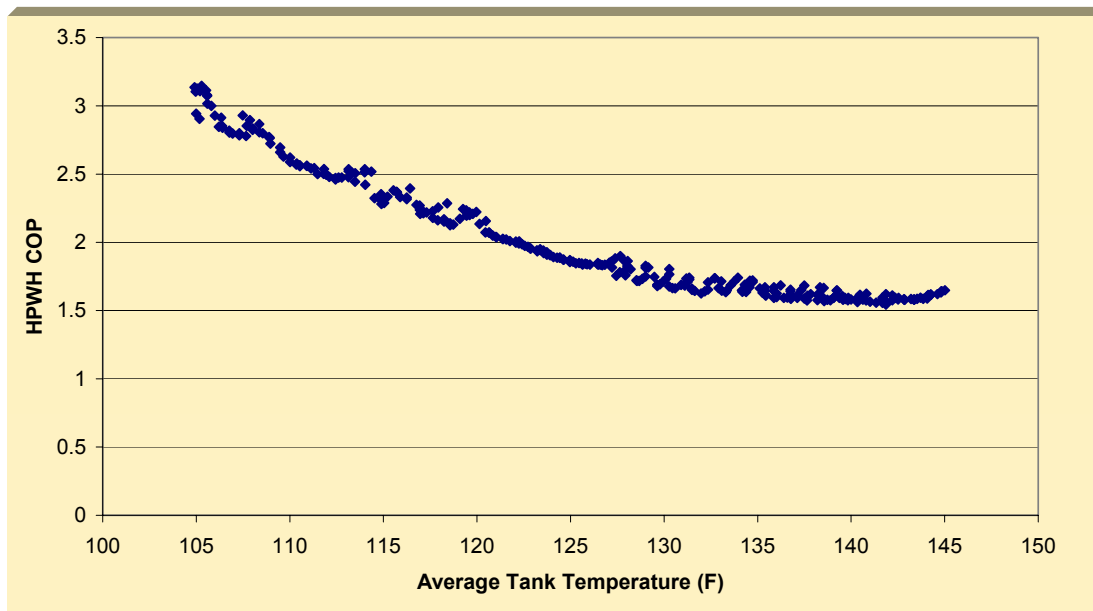
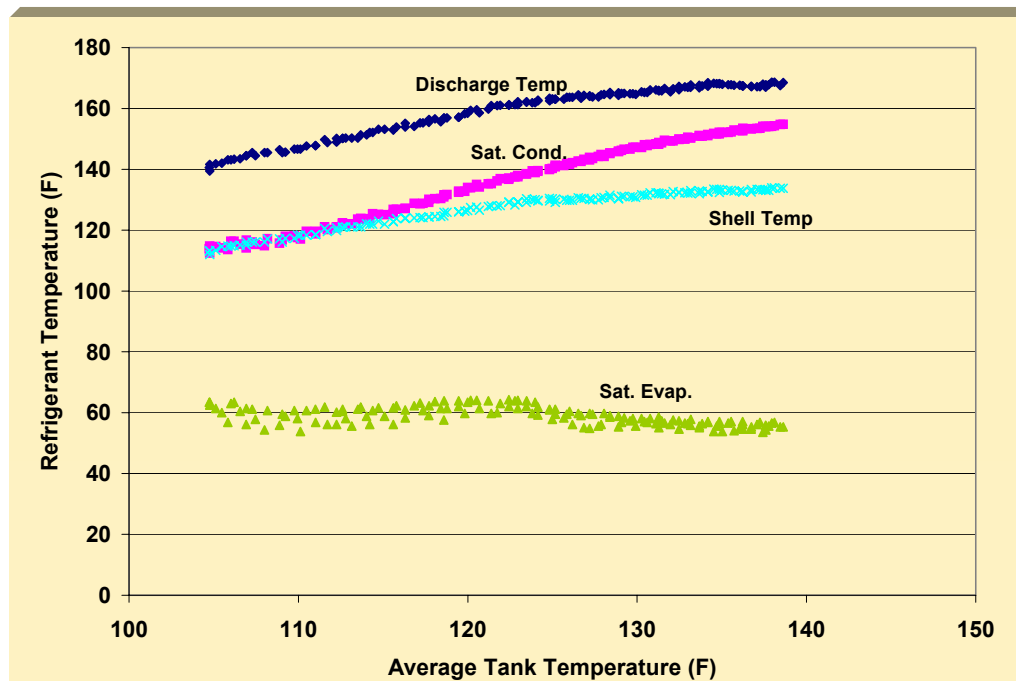**Figure D-5 HPWH COP for 50°F/80% RH Garage Test[8]**
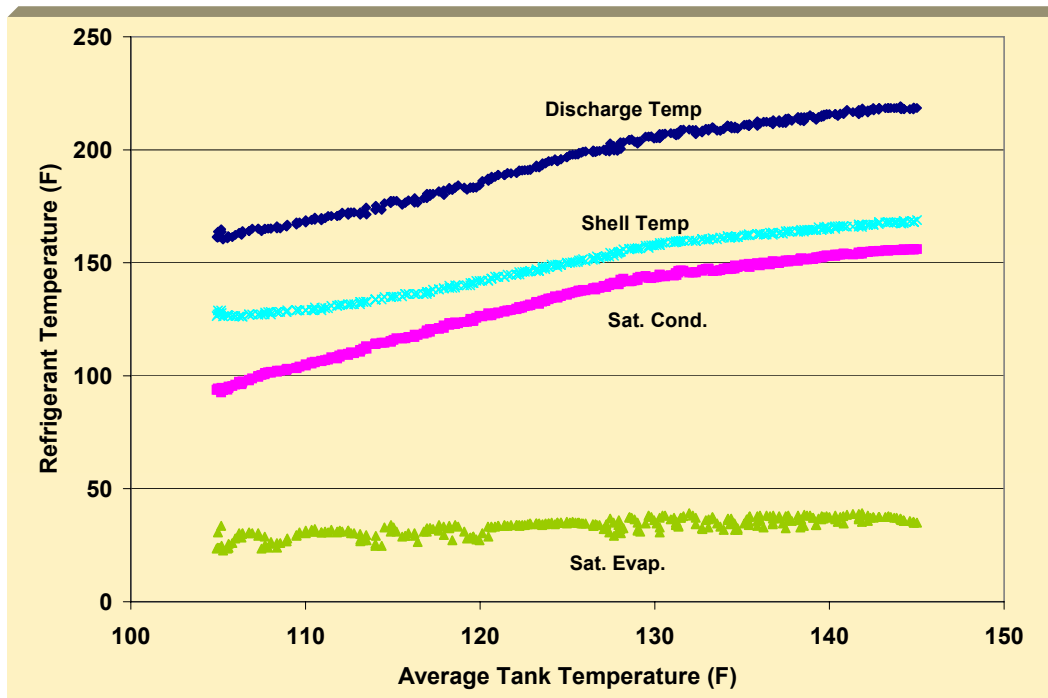
Figure D-6 Refrigerant System Temperatures for 50°F/80% RH Garage Test

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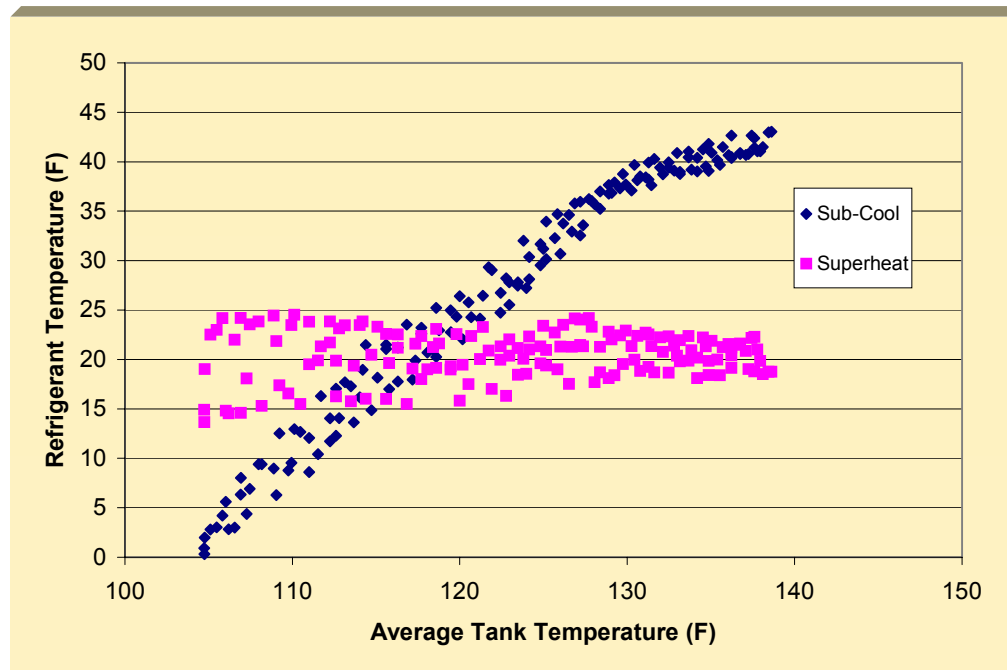[8] Calculated using raw power data and the smoothed heating capacity data.
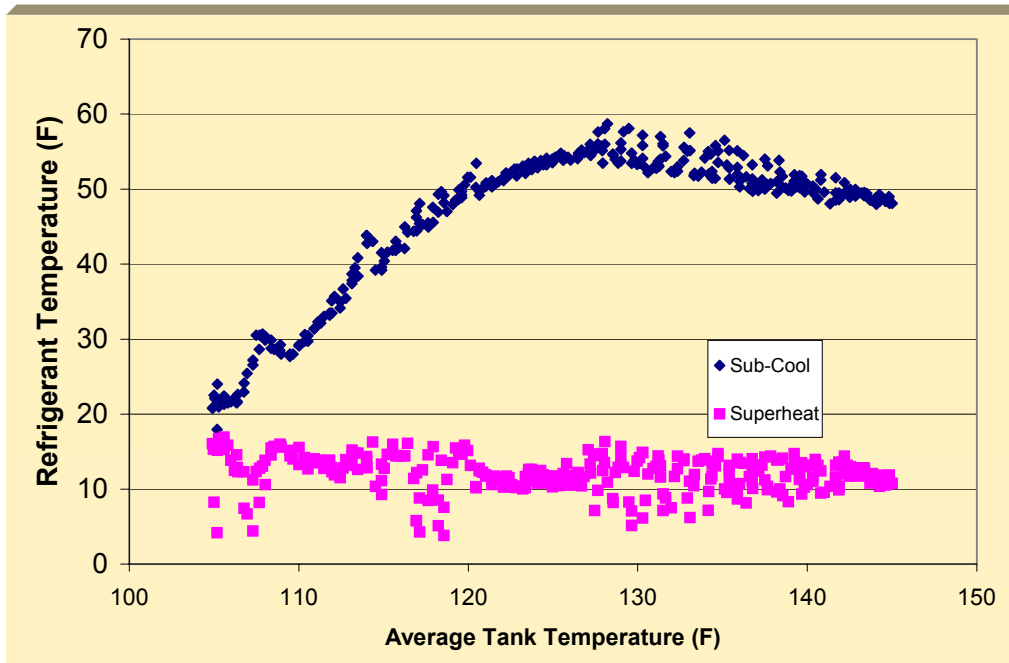
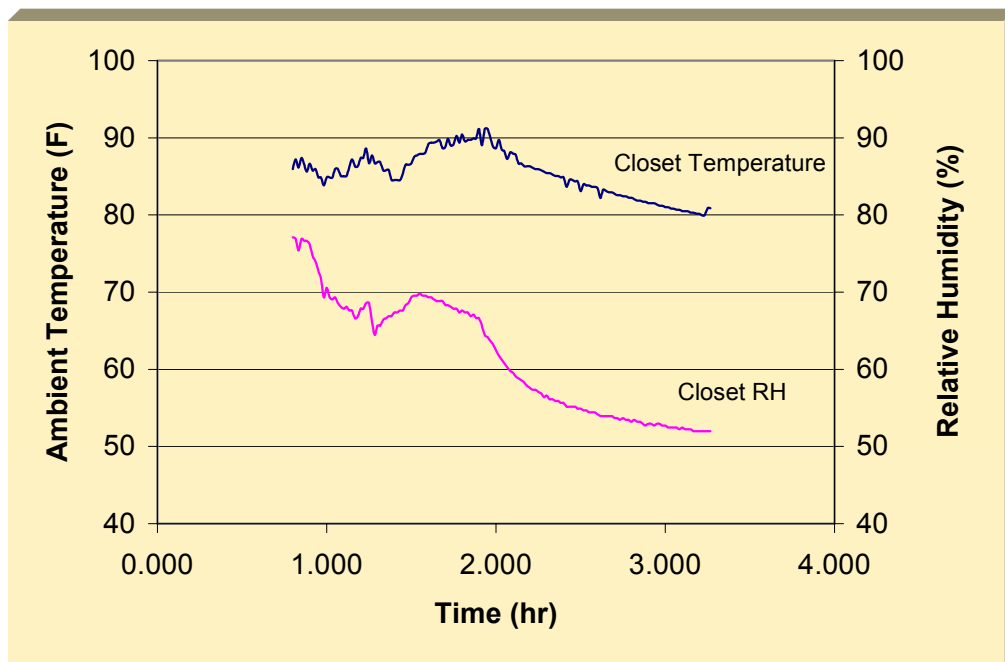**Figure D-7 Superheat and Sub-Cool for 50°F/80% RH Garage Test**
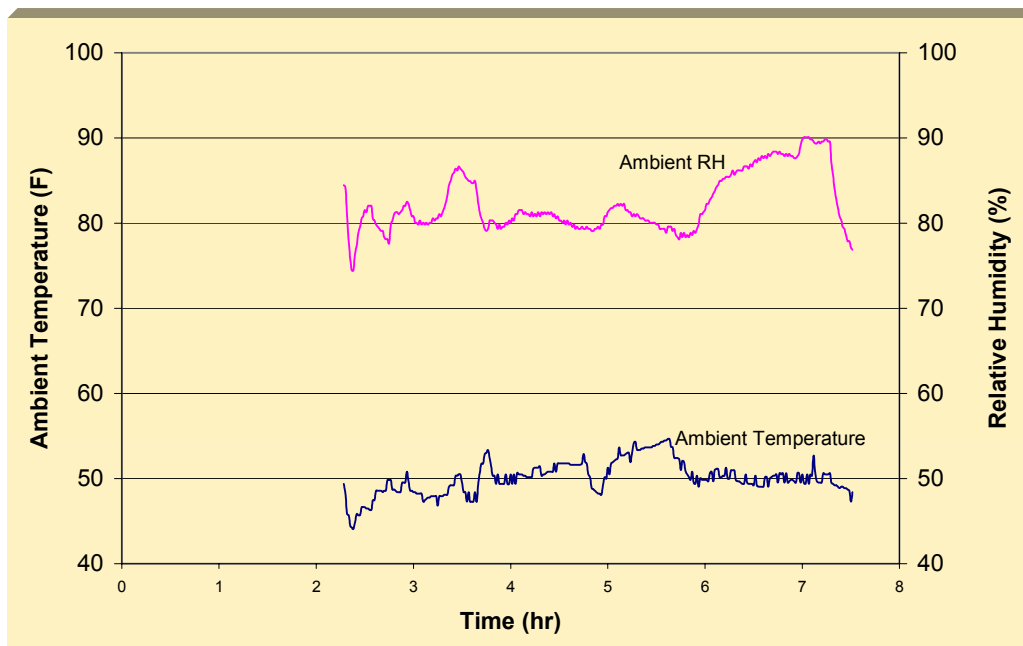
Figure D-8 Ambient Conditions During 50°F/80% RH Garage Test
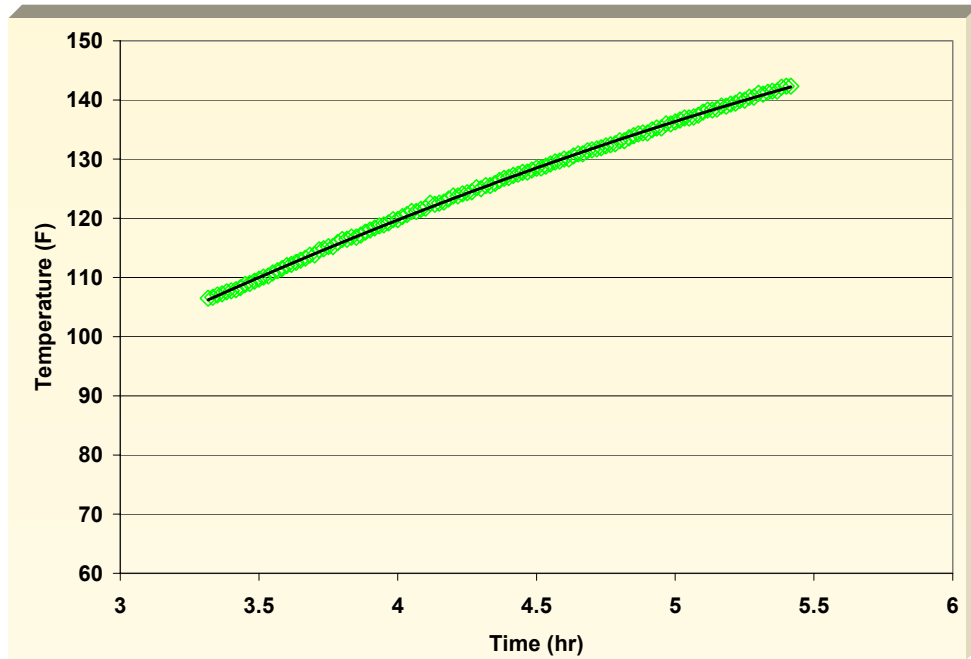
Figure D-9 Average Tank Temperature for 90°F/80% RH Garage Test
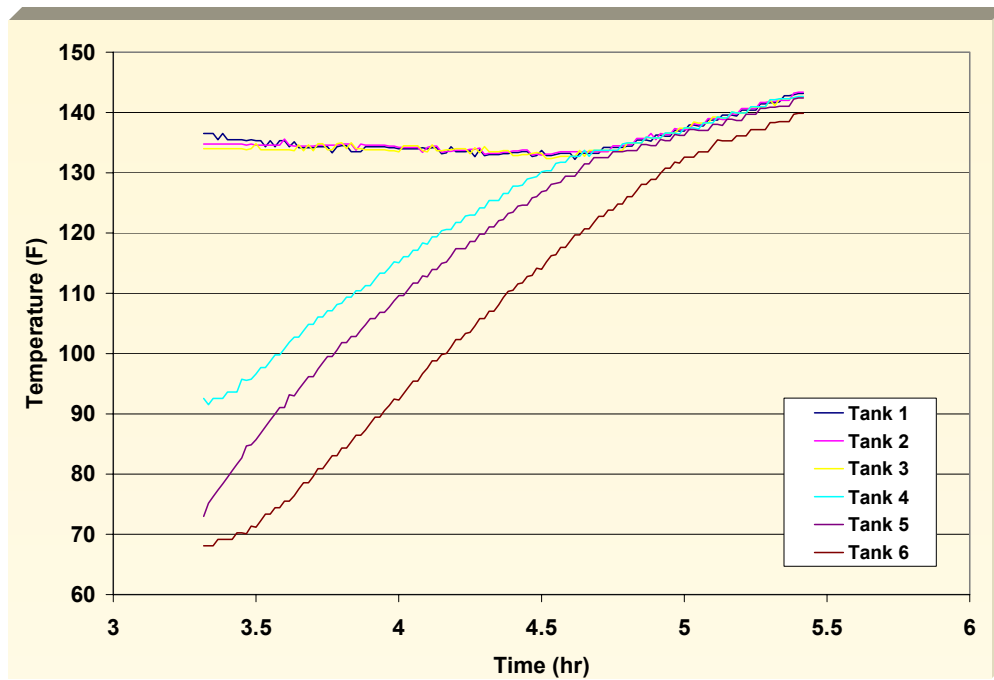
Figure D-10 Tank Temperatures for 90°F/80% RH Garage Test
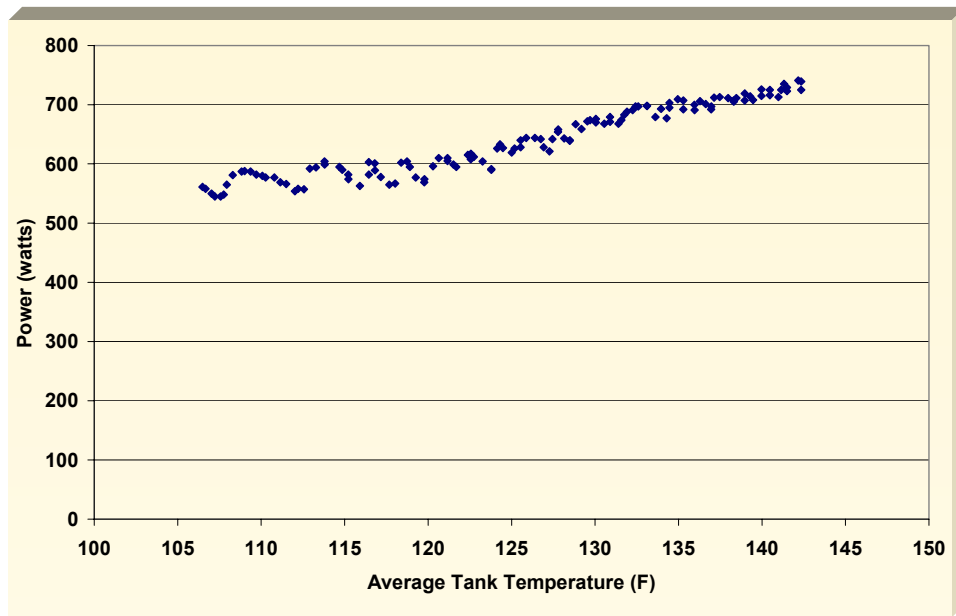
Figure D-11 HPWH Power for 90°F/80% RH Garage Test
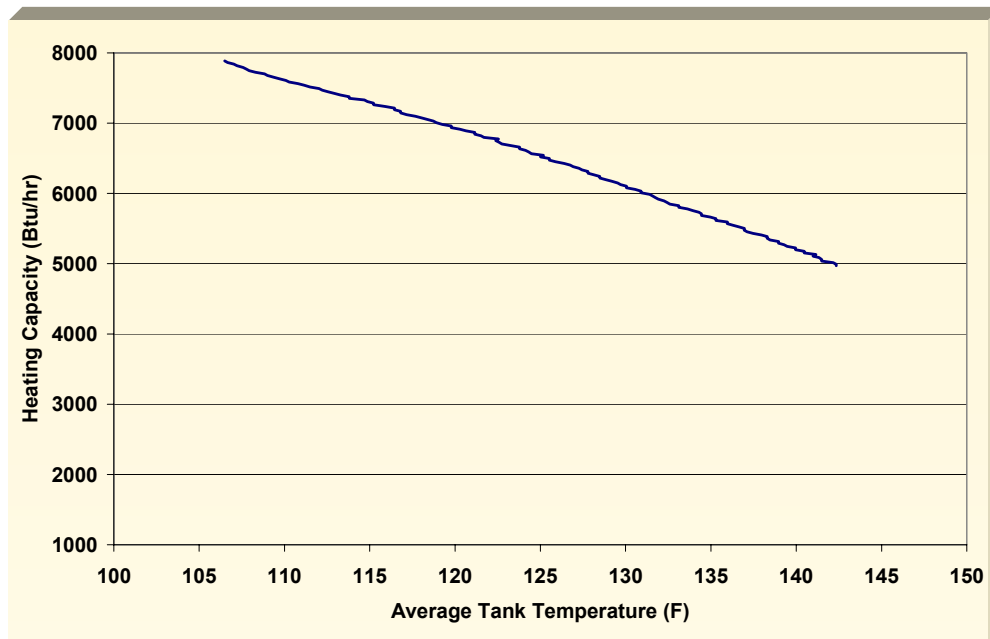
Figure D-12 HPWH Capacity for 90°F/80% RH Garage Test
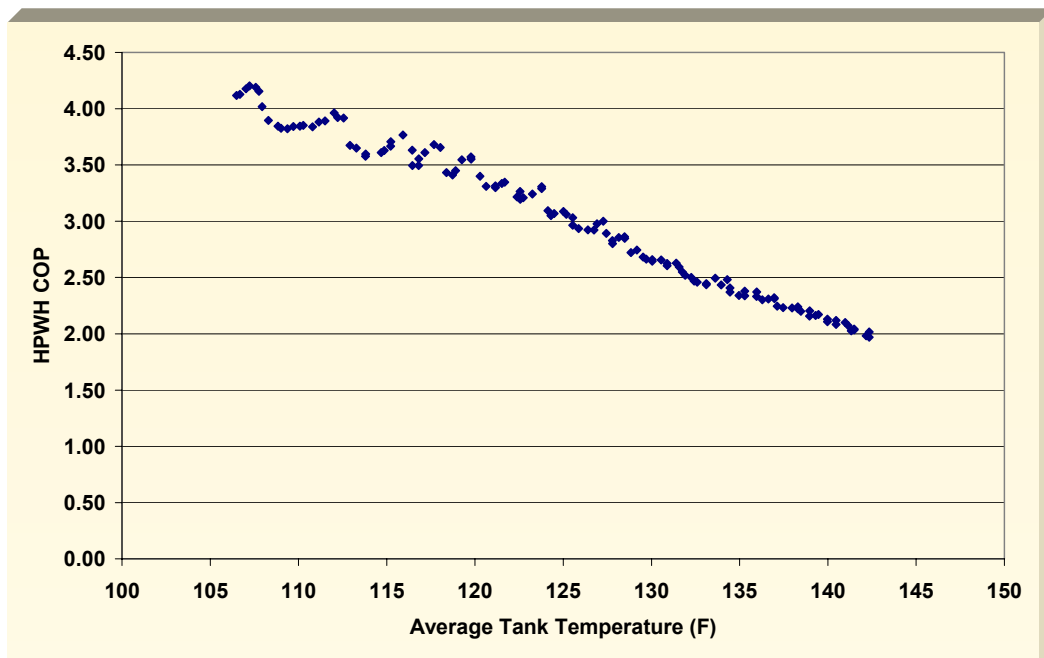
Figure D-13 HPWH COP for 90°F/80% RH Garage Test
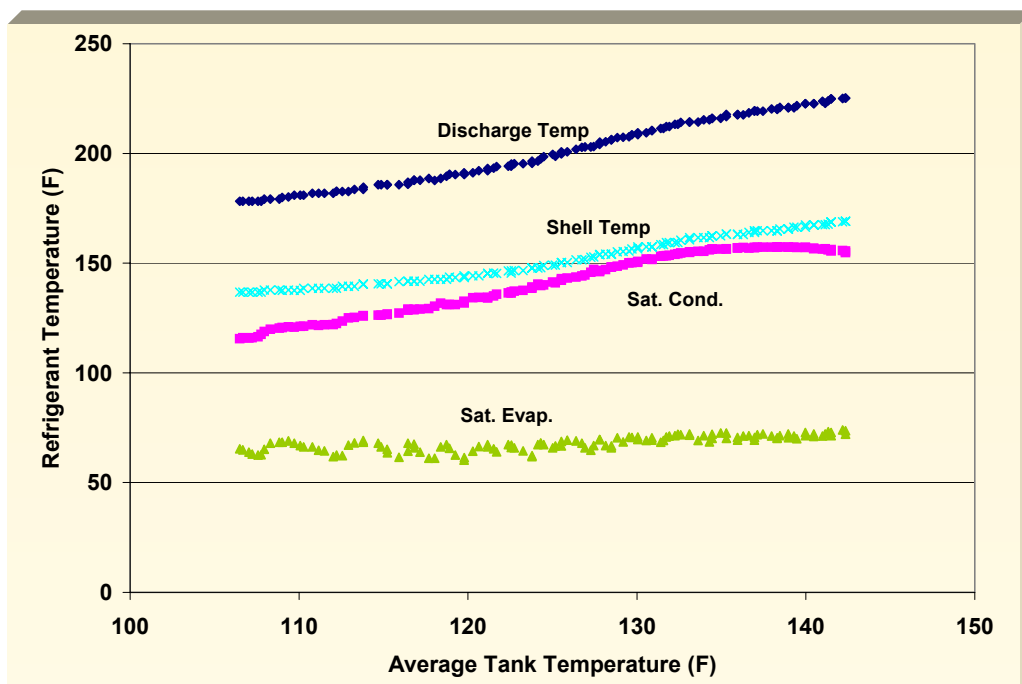
Figure D-14 Refrigerant System Temperatures for 90°F/80% RH Garage Test
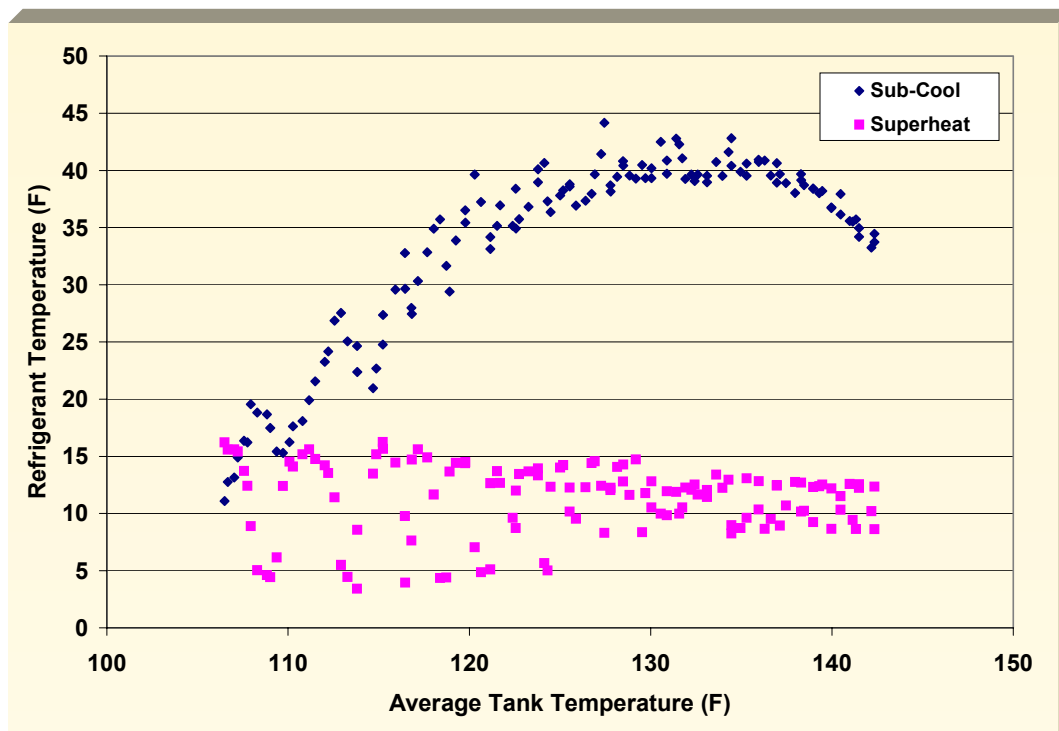
Figure D-15 Superheat and Sub-Cool for 90°F/80% RH Garage Test
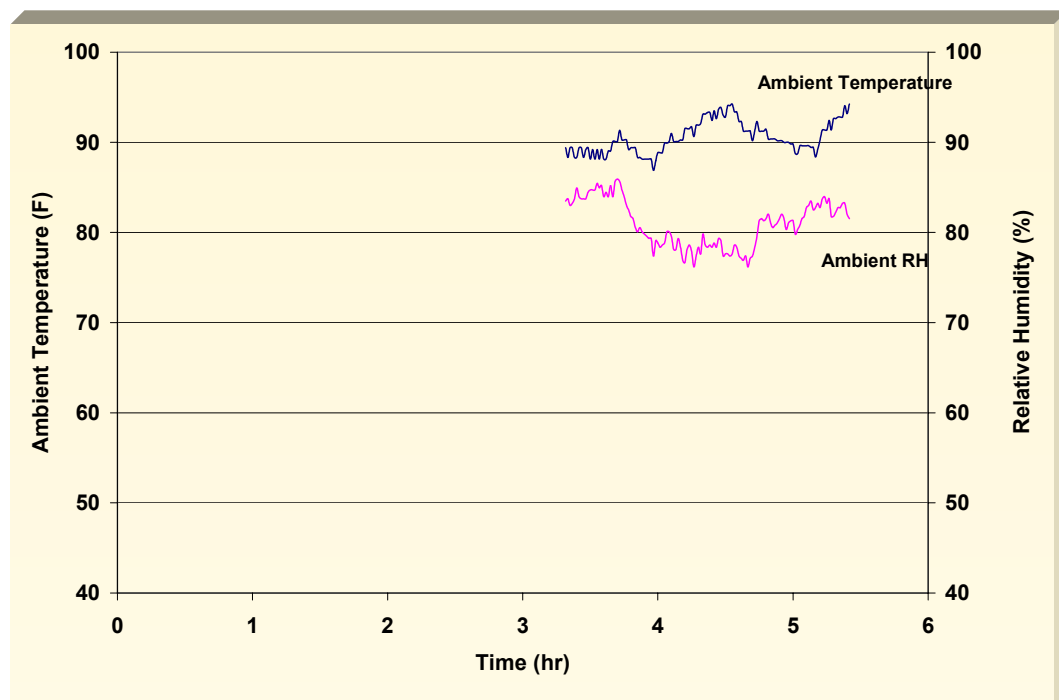
Figure D-16 Ambient Conditions for 90°F/80% RH Garage Test
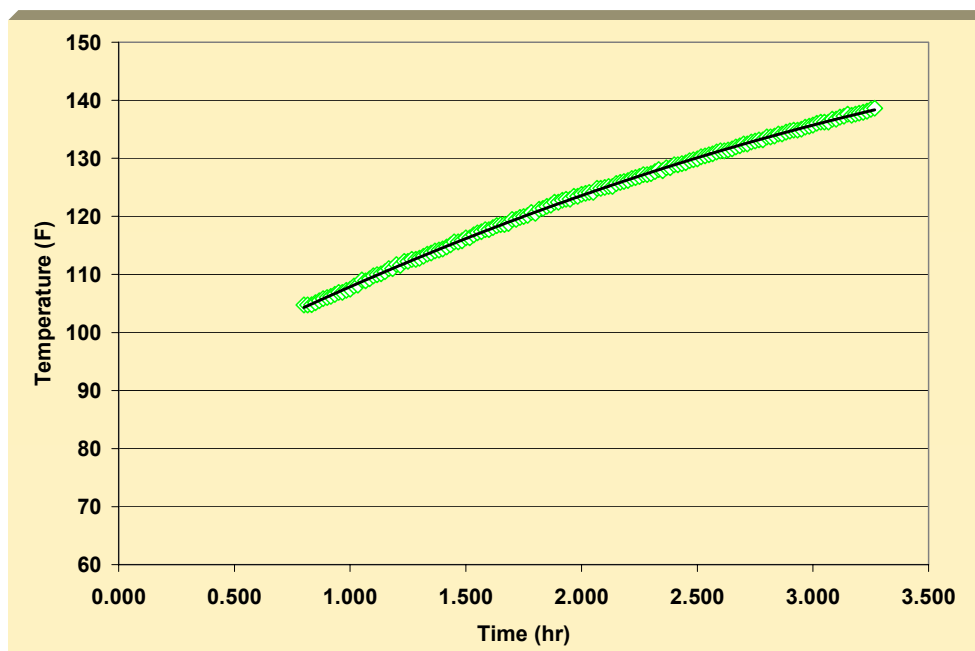
Figure D-17 Average Tank Temperature for 90°F/65% RH Garage Test[9]
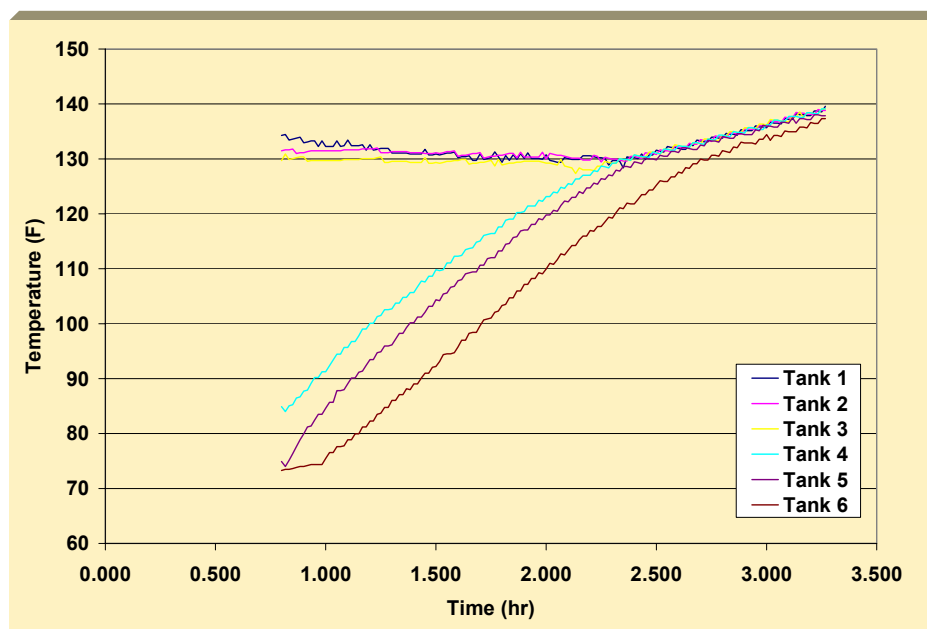
Figure D-18 Tank Temperatures for 90°F/65%RH Garage Test

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[9] The solid line represents the results of the linear regression performed on the average tank temperature
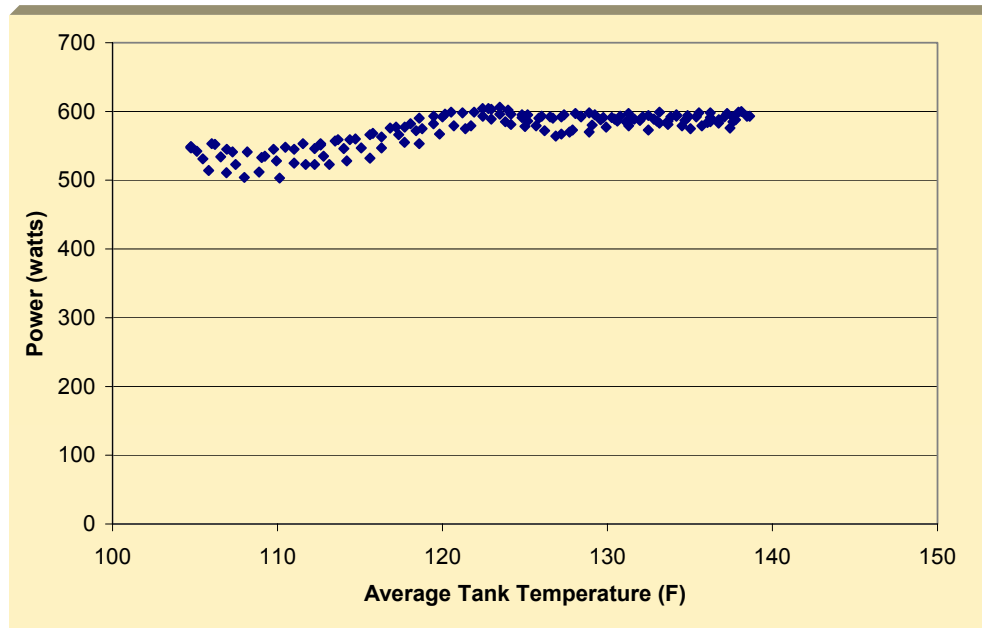
Figure D-19 HPWH Power for 90°F/65%RH Garage Test

Figure D-20 HPWH Capacity for 90°F/65% RH Garage Test[10]

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[10] Calculated from linear regression of the average tank temperature.
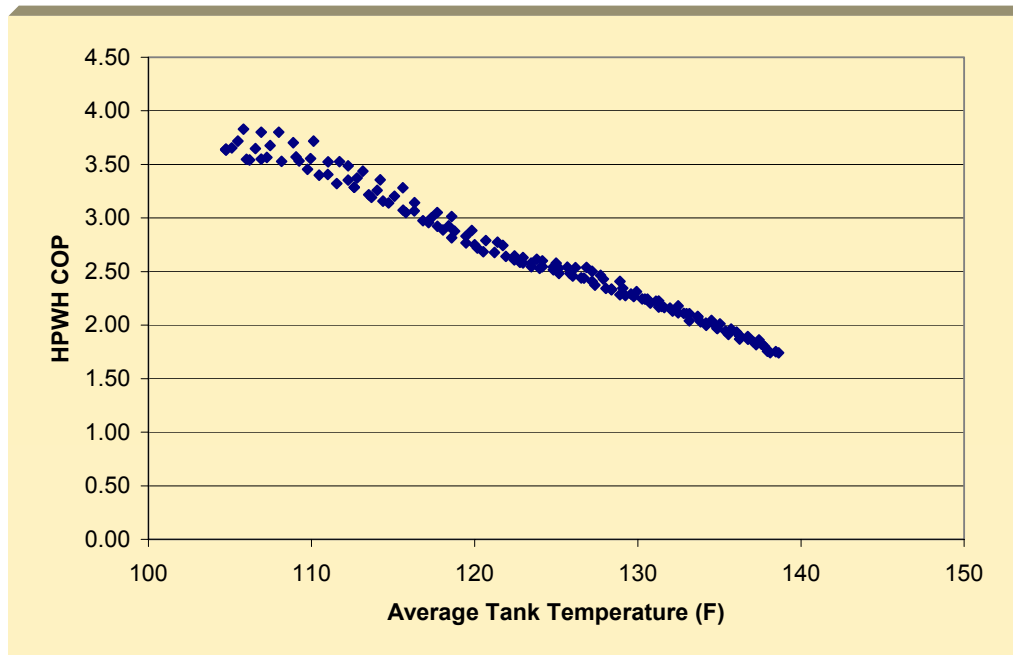
Figure D-21 HPWH COP for 90°F/65% RH Garage Test[11]

Figure D-22 Refrigerant System Temperatures for 90°F/65% RH Garage Test

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[11] Calculated using raw power data and the smoothed heating capacity data.

Figure D-23 Superheat and Sub-Cool for 90°F/65% RH Garage Test

Figure D-24 Ambient Conditions During 90°F/65% RH Garage Test

Appendix E - Closet Tests (70°F/50% RH)
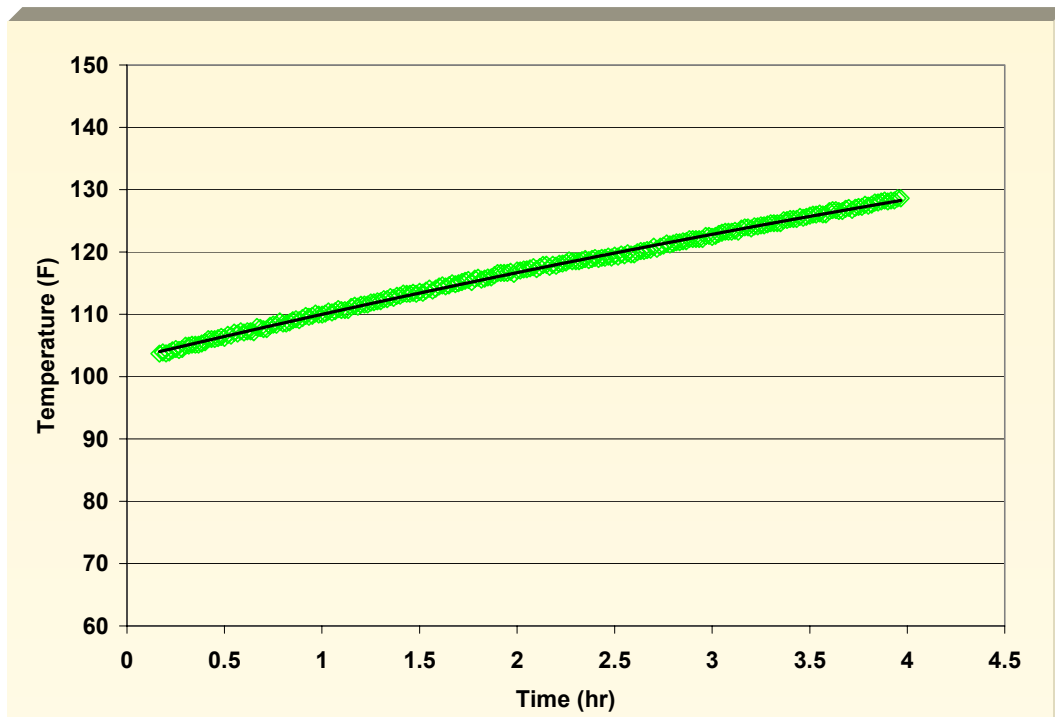
Figure E-1 Average Tank Temperature for 7°F/50% RH Closet Test
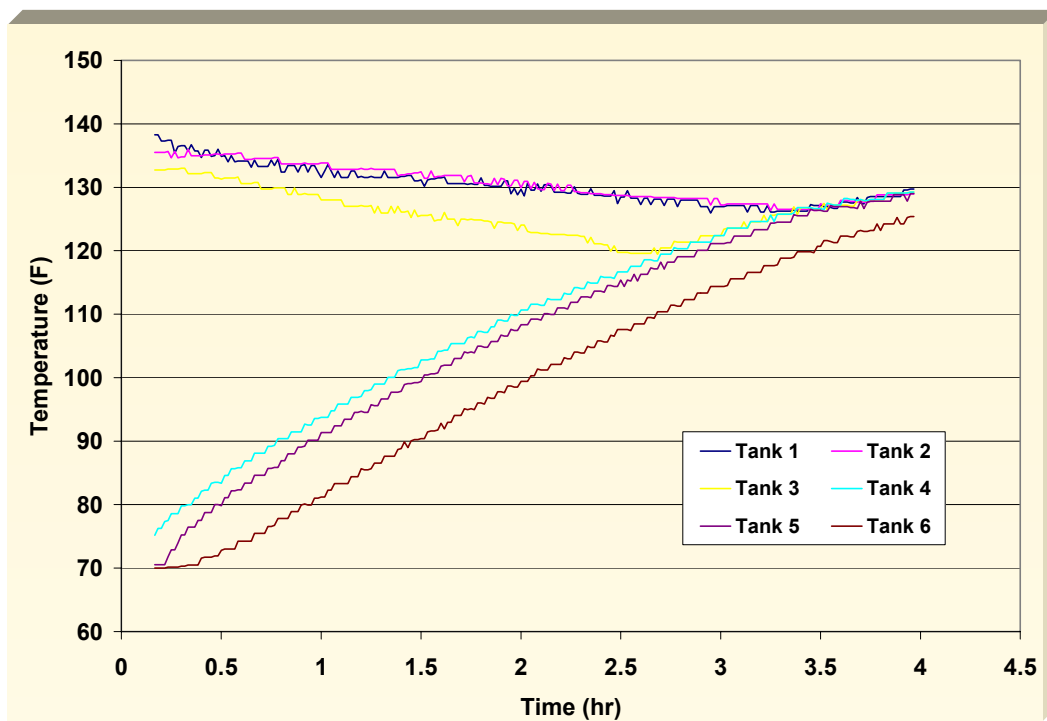
Figure E-2 Tank Temperature for 70°F/50% RH Closet Test
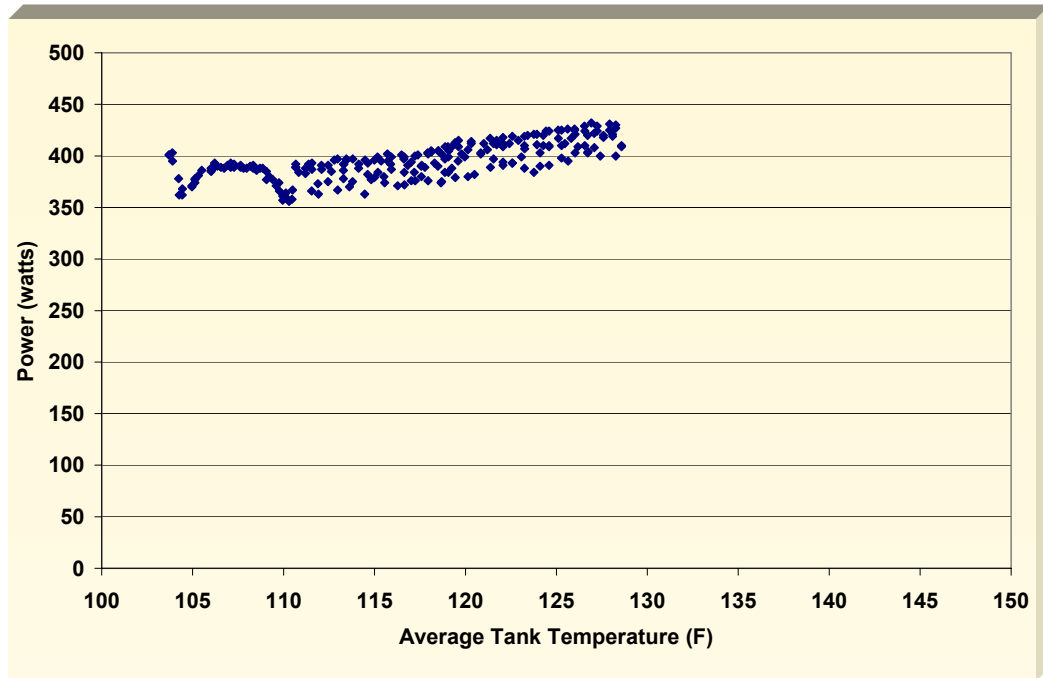
Figure E-3 HPWH Power for 70°F/50% RH Closet Test
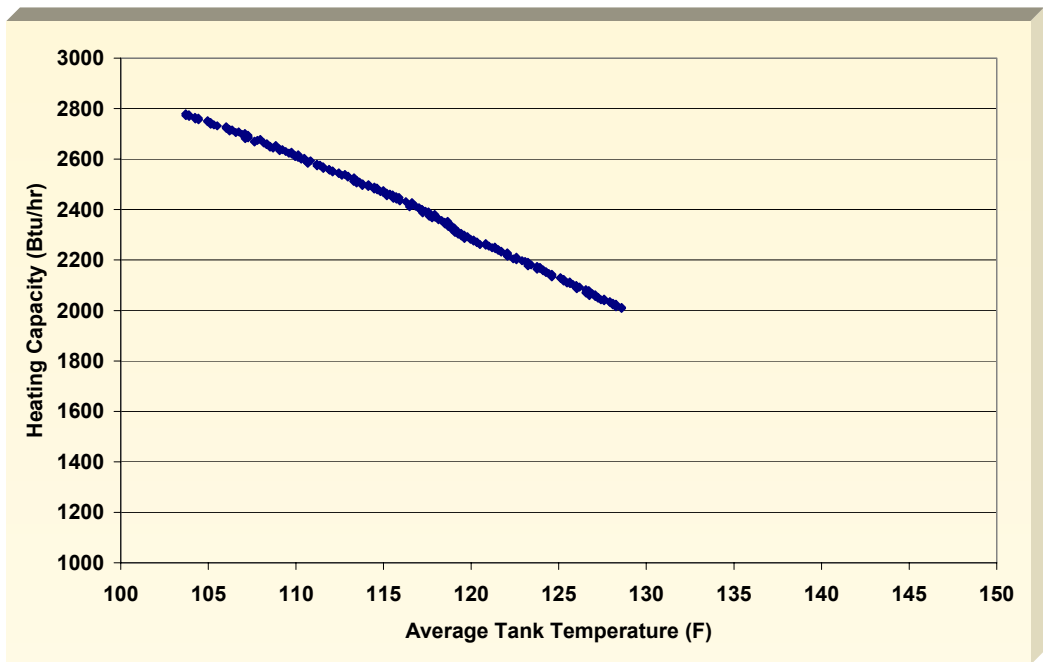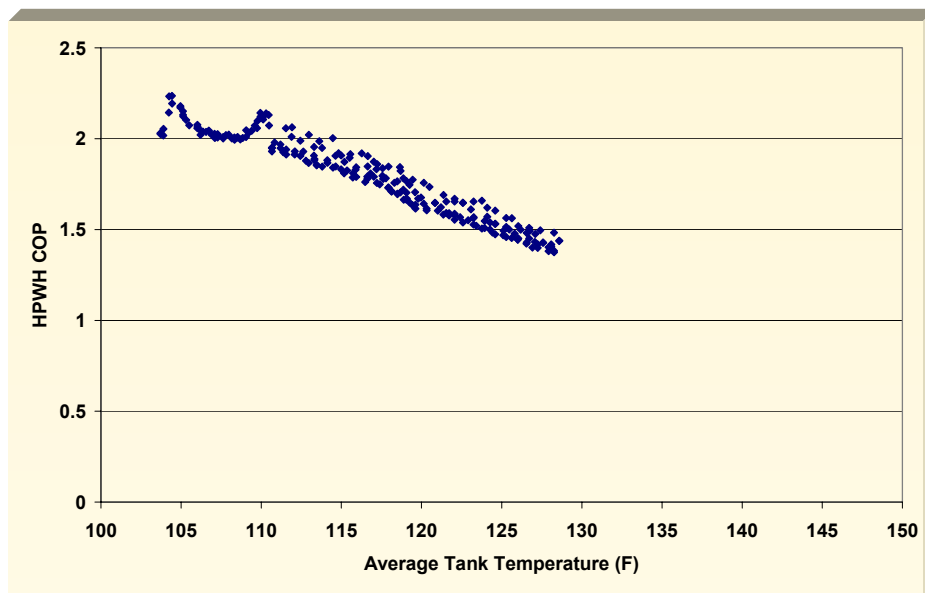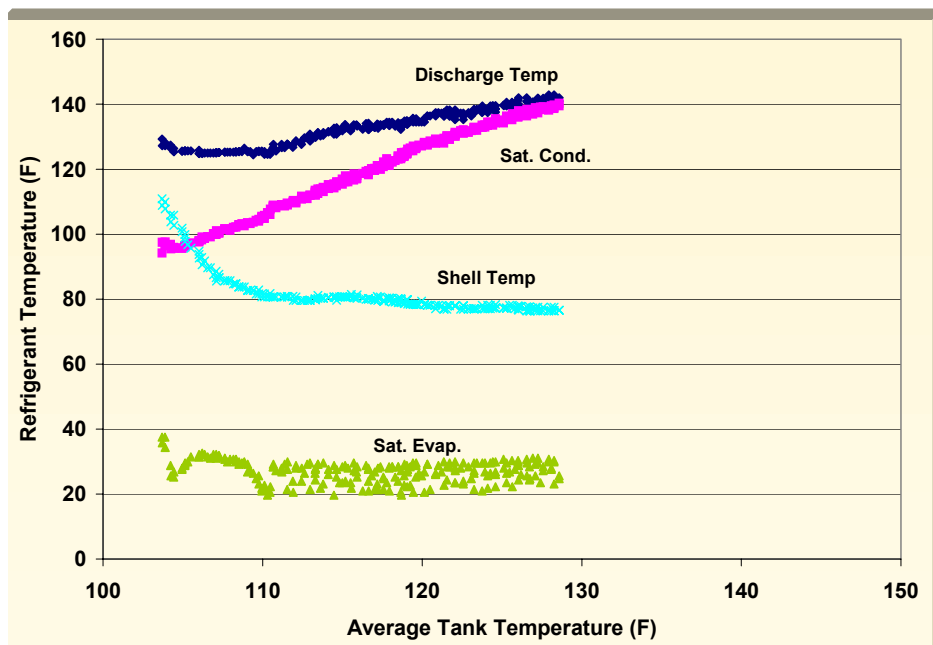

Figure E-4 HPWH Capacity for 70°F/50% RH Closet Test

Figure E-5 HPWH COP for 70°F/50% RH Closet Test[12]

Figure E-6 Refrigerant System Temperatures for 70°F/50% RH Closet Test

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[12] Calculated using raw power data and the smoothed heating capacity data.
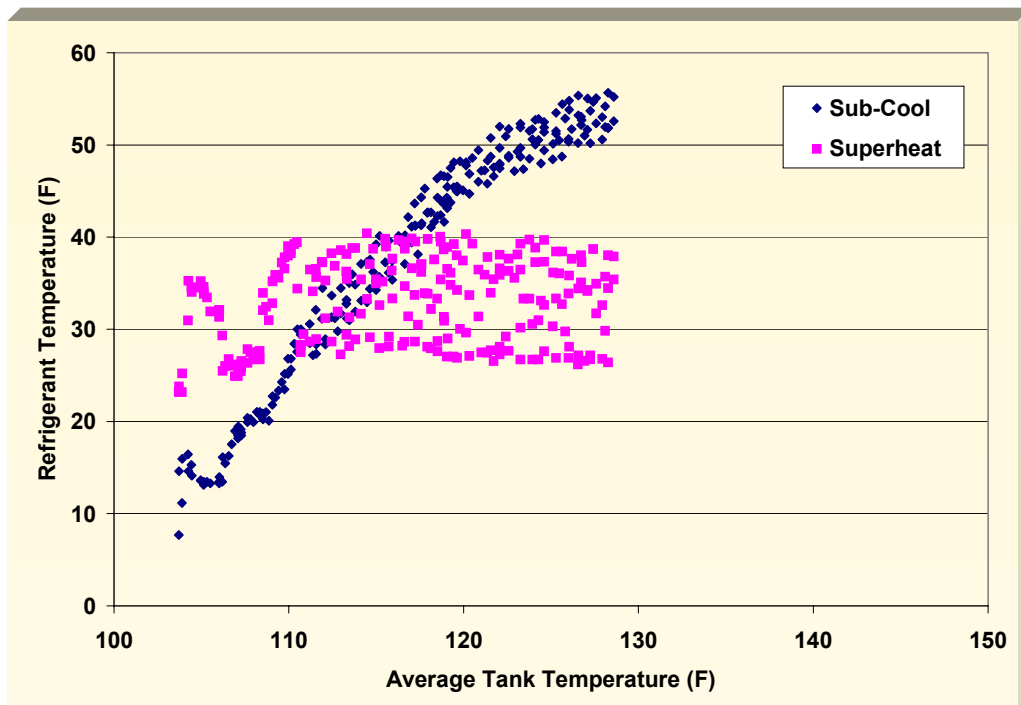
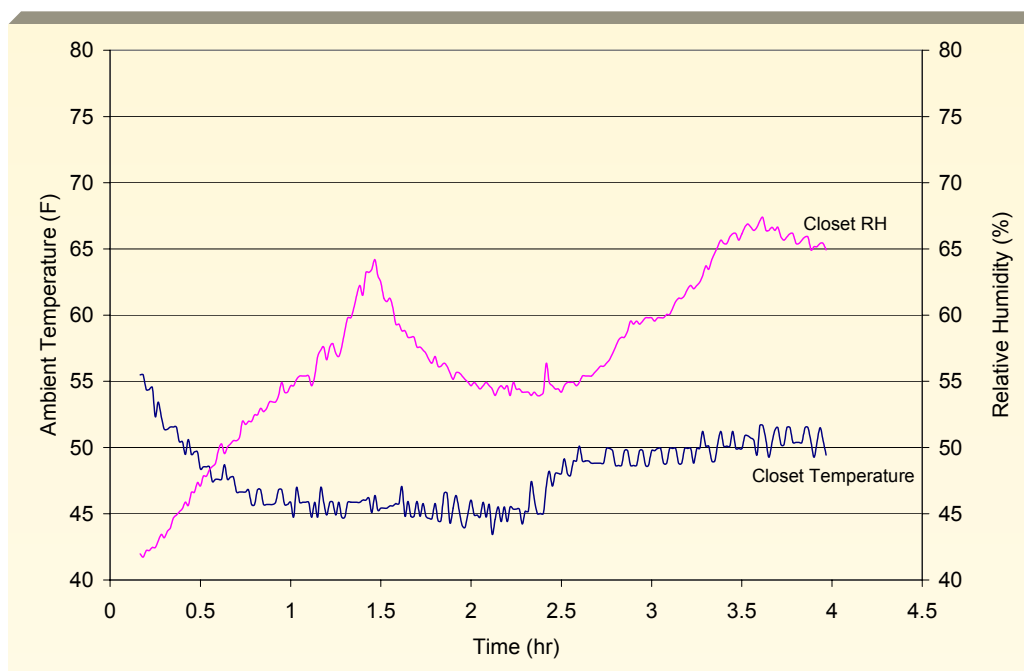**Figure E-7 Superheat and Sub-Cool for 70°F/50% RH Closet Test**

Figure E-8 Closet Conditions during 70°F/50% RH Closet Test